JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS.

VOL. IX. FEBRUARY, 1897. No. 1.

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A METHOD OF DETERMINING A CONTINUOUS RECORD OF THE PERFORMANCE OF A MARINE ENGINE.

BY PROF. W. F. DURAND, MEMBER.

The information usually derived on marine engine trials is far from furnishing data from which a continuous record of the performance can be deduced. If indicator cards are taken once in from ten to fifteen minutes, as is common on long course trials, then, so far as mean effective pressure is concerned, we have with revolutions of about 100 per minute simply one card out of every thousand or fifteen hundred. With higher revolutions we shall probably get more than one card during the time contact between the pencil and card is made, and thus the proportion may remain about the same. In the mean time, the mean effective pressure may or may not remain the same, according to circumstances. In any event, a very small sample only is obtained of what actually goes on in the cylinders. The desirability of more continuous records is well understood, and many forms of continuous integrating indicators have been proposed, though, so far, none has seemed to be a mechanical and commercial success.

As a step toward the extension of the time scope of the record, the indicator pencil is sometimes left in contact with the card for several revolutions. This, however, is of no avail beyond a comparatively small number, for although with slightly varying conditions a shaded line will result, yet we are not able by inspection to give the proper relative weight to the different parts, and to deduce a correct mean card.

In addition to the influence on the work due to varying mean effective pressure, we have also that due to varying revolutions. To obtain the mean power for a given period with the data as usually taken, two courses are open. (1) The revolutions for one minute at the time of taking each set of cards may be used with the mean effective pressure derived from them, for computing the corresponding power. The mean of the values of the power thus found may then be taken as the mean power. (2) The cards may be used simply to give the various values of the mean effective pressure from which a mean value is derived, and this may be used with the mean revolutions, as derived from the initial and final readings of the counter.

In the first case, assuming that the revolutions as counted are correct, we have still but a single minute out of 10 or 15 for revolutions and a single stroke out of 1,000 or 1,500 for mean effective pressure. It is, furthermore, too much to assume that the revolutions can be counted accurately for one minute with the ordinary means. Unless the observer is experienced and uses great care, the error will often rise to as much as one per cent. or more, while fractional parts of a revolution can not be satisfactorily approximated to in this way. The objection to the second method lies in the fact that the use of a mean value of the revolutions with a mean value of the mean effective pressure, even if the latter were correct, would not give a correct mean value of the power. The error is usually quite small; still it is none the less real, and its cause may be properly considered at this point.

Work is proportional to the product of two factors—the force factor and the distance factor. The former is represented by the mean effective pressure and the latter by the revolutions. The

force factor is moreover directly proportional to the ship resistance, and the latter to the nth power of the speed, where n is usually from 2 to 2.5, approaching or even exceeding 3 at critical or high speeds. It follows that we may represent work in the form:

$$W = A R^n R = A R^{n+1}$$

Where A is nearly constant for moderate changes in the conditions, R^n represents the force factor and R the distance factor. Now, with all such algebraic forms it is a well known fact that with varying values of R, n not being 1, the mean of the nth powers of R multiplied by the mean value of R will not be the same as the mean of the $(n+1)$st powers of R. This may be illustrated by a simple and extreme case.

Let R vary continuously from 0 to R, and let $n = 2$. Then by integration we have for the mean values of R^2 and R, $R_1^2 \div 3$ and $R_1 \div 2$. Hence the product would be $R_1^3 \div 6$. The true mean value of the work will be represented by the mean of R^3, and this is $R_1^3 \div 4$. In such case, therefore, the value derived by multiplying the mean pressure by the mean revolutions would be less than the true mean in the ratio of 4 to 6. This is a range of variation, of course, entirely beyond anything we are apt to meet with in engine trials, and is simply of interest as showing the extent to which such an error might rise. If we suppose more generally that the revolutions vary continuously from R_1 to R_2, then we shall have for the ratio between the two values:

$$\frac{2}{3} \cdot \frac{(R_2 + R_1)(R_2^3 - R_1^3)}{R_2^4 - R_1^4}$$

If R varies continuously from 80 to 100 the resulting value of the ratio is about .992, or the error would be nearly one per cent. With a corresponding variation of R from 90 to 100 the value is about .998, and the error about one-fifth of one per cent. With values of n higher than 2 the errors would be relatively greater. With only moderate variation in the revolutions, however, it is readily seen that the error is quite within the usual range for engineering work of this nature.

Of these various sources of error in the determination of the mean power, that due to the variation of the mean pressure is apt to be the most important, and more or less so as the pressure is more or less irregularly variable.

A continuous record of test data would, moreover, be of much value for many purposes other than the more correct determination of the mean power. Many questions likely to arise in the detailed analysis of a test could be much more intelligently studied from data in this form than in the form usually found. We may, therefore, omit further discussion of the desirability or uses of such records and proceed to a consideration of the elements involved in them and of ways and means of making them continuous.

We take first the mean effective pressure. This depends on (1) the initial pressure, (2) the vacuum, (3) the point of cut off and other characteristics of the engine. Now as tests are usually run, the cut off remains fixed throughout the entire time, and for the present, at least, we may consider that (3) remains constant for the entire test. We are therefore left with (1) and (2). If then we can determine experimentally any general relation connecting the mean effective pressure with the initial pressure and vacuum, and can then obtain a continuous record of the latter, we shall have the means of determining a continuous record of the former. The relation sought must be determined by experiment and consists in fact of a calibration or standardizing trial, in which the steam pressure and vacuum will be determinately varied, and indicator cards will be taken. Suppose for example that the working pressure is 150 pounds by gauge, and that about 25 inches of vacuum can be carried. Then suppose the ship to steam out in free route, and let the pressure be varied from say 135 or 140 to 160 by 5-pound increments, the vacuum being kept as nearly as possible constant at the 25 inches. Cards are taken from all cylinders when the initial pressures are at the desired points, or at least observations of pressure are made simultaneously with the taking of the cards. For proper assurance these observations should doubtless be repeated, so that two series of cards would be obtained, one on a rising and one on a

falling pressure. Then let this work be repeated at a vacuum of about 20 inches, lowered by slowing the air or circulating pump, or by the admission of air into the condenser. This will comprise all ordinary range of variation for vacuum, and will complete the observations for this point of cut off. The cards being measured up, preferably of course by planimeter, will give values of the mean effective pressure for each end of each cylinder, which can be plotted so as to give lines showing the continuous relation between initial and mean effective pressures, each for a single value of the vacuum. By interpolation we may then readily determine the value of the mean effective pressure corresponding to any value of the initial pressure and vacuum contained between the limits of the observations.

In order to make use of this information for the determination of mean effective pressure, we have on the actual trial simply to provide for the continuous record of the steam pressure and vacuum. This may be most readily done by recording gauges, or, if they are not available, by readings taken by an observer at, let us say, quarter-minute intervals. Such readings give to all intents and purposes a continuous record, and will answer quite as well as the record from a recording gauge. In any case, of course, the gauge which gives the continuous record should be the same as that from which the pressures were read on the preliminary or calibration trial. It may also be noted that it is of no particular importance whether or not this gauge is accurate in its indications, since its only purpose is to enable us to connect conditions at any given instant on the actual trial with the mean effective pressure as derived from the cards taken on the calibration trial. The gauge should, of course, be sensitive and constant under constant conditions, but need not be correct absolutely in its indications.

In a similar manner a continuous record of the vacuum should be determined.

Having thus a continuous record of the steam pressure and vacuum, and a continuous relation between these and the mean effective pressure as determined by calibration, we may determine a continuous record of the mean effective pressure. The assump-

tion here implicitly made is, of course, that the condition of the engine is the same, or at least that all influences which may affect the mean effective pressure are the same as on the calibration runs. We shall recur to this at a later point.

We have next to consider ways of obtaining a continuous record of the revolution per minute. For this there are forms of recording tachometers which might answer the purpose. In default of such instruments we may instead read the counter every quarter or half minute, putting down the numbers in succession, but making no attempts at subtraction to find the differences. Here again, as we shall show later, absolute accuracy is not so desirable as relative correctness of value. With correct relative values and the true total integration of the revolutions, as given by the counter, we shall see how the record derived may be made correct both relatively and absolutely. This being done, we shall have continuous records of both mean effective pressures and revolutions, and hence by their combination a continuous record of the work and power in each end of each cylinder, and hence by summation a continuous record for the power as a whole.

Having thus discussed the method in the abstract, it may be of interest to proceed to the examination of the data and results of a trial made in this way. The trial was made on Cayuga Lake during the past summer under the writer's immediate supervision, the vessel used being the steam yacht *Clara*, belonging to Mr. Charles Kellogg, of Athens, Pa., through whose kindness the experiment was made possible. The principal dimensions of the *Clara* are as follows:

Length, feet,	91
Beam, feet,	11.9
Draught, feet and inches,	3.5
Displacement, tons,	40

The engine is compound, the cylinders being 10 inches and 19 inches in diameter by 12 inches stroke.

Calibration runs were made at steam pressures by the boiler gauge of 70, 80, 90, 100, 110 and 120 pounds for two positions of the link, and for nearly constant vacuum at 23.5 inches. Two

cards were taken for each set of conditions, an ascending and a descending series, thus making in all 24 sets of cards. This formed the calibration data. To show its application, a short trial run of 30 minutes was then made, in which readings were taken every 15 seconds for steam pressure, vacuum and counter. No cards, of course, were taken during the latter run or actual trial. The steam pressure and vacuum were allowed to freely vary, the range of the former being about 30 pounds and of the latter about 2 inches.

The cards taken during the calibration run were then worked up for mean effective pressure. The results for the cut off used on the trial are shown in Figures 1 and 2. These will be readily understood without special explanation. The results for the other cut off are quite similar, and need not be repeated.

As might be anticipated, the relation between the initial pressure and the mean effective pressure is practically linear. This naturally follows from the fact that, except for thermodynamic interactions, the mean forward pressure must vary directly with the initial pressure, and with the general combination of valve events and expansion ratios. But the latter are designedly left constant by a fixed position of the link, so that except for secondary modifications due to different thermodynamic relations between the mean forward pressure and the initial pressure, the former should be a linear function of the latter. The same is true for the back pressure except for the influence of the condenser pressure, which is supposed to remain constant. Hence, except for thermodynamic reasons, the mean effective pressure should vary as a linear function of the initial pressure. This is borne out by the diagrams, the values for the most part falling very close to a straight line and showing no conclusive tendency toward a curvature in either direction. Whether this linearity of law would hold more generally, or for a wider range of pressures, is a point admitting of doubt. For a moderate range of pressure, however, there seems no reason to question the practical linearity of the law. This being accepted, the work of calibration would be very much reduced, two or three points well determined being all that would be necessary. Another point

plainly shown was that the high pressure mean effective pressure was practically independent of such variation as occurred in the vacuum, the range of variation being about 2 inches. This may also be readily understood when we remember that the only influence of the condenser pressure on the high pressure cylinder is that due to the steam shut into the low by the closure of the exhaust. This influences the receiving line of the L.P. card and hence the back pressure of the H.P. card. In amount, however, this influence is so small as to be within and obscured by the necessary observational uncertainties attending all such work. It was also shown that for a range of vacuum variation not greater than that above mentioned, viz., 2 inches or about 1 pound, the effect on the L.P. mean effective pressure was simply addative or subtractive by the amount of the variation. This also is readily seen to follow from the consideration that a change in the vacuum means practically a raising or lowering of the back pressure line all along by the same amount. Actually this difference is increased on the compression line, but the influence of this is too slight to practically affect the simplicity of the relation for slight changes in the vacuum.

For wider changes it would doubtless be preferable, as previously suggested, to make separate runs at a considerably reduced vacuum in order to have the influence of this variation strongly marked. The results of the calibration are therefore plotted as shown in the figures. For the H.P. one line is sufficient; for the L.P. three are shown for varying vacuum, thus making it simple to obtain by interpolation the value for any initial pressure and any vacuum within the range of the observations.

We pass next to the data taken on the 30-minute run representing the actual trial. The steam pressure and vacuum were read as above explained, and the results are shown plotted on a time abscissa in figure 3. In this diagram, $C\ O$ is the record for the steam pressure and $E\ F$ that for the vacuum. With regard to the data for revolutions, an inspection of the counter readings showed that the best mode of treatment would be the following.

The first reading was subtracted from the fifth, or from that taken one minute later. This difference was considered as giving

a value of the revolutions for the first minute. This number was then plotted as an ordinate at the abscissa ½ minute, or, in other words, at the middle of its range. Reading number 2 was then subtracted from number 6, and the result was taken as the number of revolutions for a minute located 15 seconds later than the first. This value was plotted at an abscissa ¾ minute, or 15 seconds beyond the first point. In like manner the difference between readings 7 and 3 was plotted on abscissa 1 minute, and so on. The result was a series of numbers of revolutions per minute for minutes located 15 seconds apart along the scale of abscissa. The result gave a fairly continuous curve, with here and there a few scattered points due to the necessary errors of observation in such cases. The mean path through such points was, however, clearly indicated by the general direction of the curve on either side, and there was no difficulty whatever in drawing a line which was clearly a very close representation of the law of the revolutions. This is shown at *A B*.

While the ordinates of such a curve might be quite correct relatively, they might, nevertheless, be somewhat in error absolutely. In order to test the latter point the following method was used. The integration of the curve would, of course, furnish a mean value of the revolutions for the entire time interval. The true mean revolutions, however, would be given by the total integration as furnished by the counter, divided by the number of minutes in the total time interval. If these two values closely agree, we may conclude that the curve derived as in *A B*, figure 3, gives values substantially correct both relatively and absolutely. If not, its relative indications may still be taken as the most reliable to be obtained under the circumstances, and the whole series of ordinates of the curve may be multiplied by the ratio between the true mean revolutions and the value derived by the integration of the curve. We may then consider that the results will be correct both relatively and absolutely to as high a degree as can be derived from the data in question.

In the present case, the first value was derived by integrating the complete set of ordinates at 15-second intervals by parabolic rule. The result was 228.36 revolutions per minute. The true

mean, as derived by subtracting the first counter reading from the last and dividing by 30, gave as a result 228.46. The agreement is remarkably close, the difference being less than one part in 2,000, and therefore falling far below significance in engineering work of this character. We may therefore safely conclude that the curve *AB* thus derived represents to a high degree of accuracy the continuous history of the revolutions throughout the run. The close agreement in general character between the curves for initial pressure and for revolution may be noted, the slight variation in the latter being probably due to wind puffs or to other accidental causes.

This point being reached, the remainder is simply a matter of detail.

The mean effective pressure for each end of each cylinder is read off, and if desired may be plotted as in figure 4. These, combined with the revolutions, give the work in each end of each cylinder. This is shown in figures 5 and 6. Finally, by addition, the total work is found, and this is shown plotted in figure 7. In these latter or final resultant curves, the values are plotted at minute intervals, a continuous smooth curve through such points giving all significance of representation. It is not necessary, of course, to plot, as in figures 4, 5 and 6, the curves of mean effective pressure and of work in each end of each cylinder. They are here shown in order to fully illustrate the successive steps from first to last. As an additional point, it may be mentioned that the actual diagrams are plotted on the usual cross section paper, in which each of the divisions shown in the diagrams is divided into ten parts. For simplicity the principal divisions only are shown here, as they are sufficient to illustrate the principles involved. As a final point, the power curve of figure 7 was integrated and the value of the mean power was found to be 105.23.

A comparison between this result and that given by individual cards taken at various intervals will not be without interest. Taking each card with its own revolutions, we should have the following results:

With intervals of 15 minutes, the first card being taken 7½

minutes after the beginning of the run, we have 115.3 as the mean power.

With intervals of 15 minutes, the first card being taken at the beginning of the run, we have the value 95.7.

With intervals of 10 minutes, the first card being taken 5 minutes after the beginning of the run, we have 109.2.

With intervals of 10 minutes, the first card being taken at the beginning of the run, we have 102.6.

With intervals of $7\frac{1}{2}$ minutes, the first card being taken $3\frac{3}{4}$ minutes after the beginning, we have 109.

With intervals of $7\frac{1}{2}$ minutes, the first card being taken at the beginning, we have 103.4.

With intervals of 5 minutes, the first card being taken $2\frac{1}{2}$ minutes after the beginning, we have 109.2.

With intervals of 5 minutes, the first card being taken at the beginning of the run, we have 105.4.

The closeness of the last result to the true value is quite accidental, as may be seen by noting that with the same time interval, differently located, the result is 109.2. It is not, in fact, until we reach a time interval of 3 or 4 minutes that we can depend upon an error not greater than about one per cent. If, instead of using with each of the above selected values of the mean effective pressure the corresponding value of the revolutions, we should find from them a mean mean effective pressure, and use this with the true mean revolutions as deduced from the counter, we should find somewhat more accurate results. Still, with this method, it is not until an interval of about 5 minutes is reached that the error is reduced to about one per cent. It may be noted, however, that the relative fluctuation of the conditions on the above trial was greater than would be experienced in any ordinary case. This would, of course, decrease the probable error of the results in comparison with that illustrated above. Cards can, of course, be taken at 4 or 5-minute intervals if a sufficient number of observers are provided, but in such case the labor involved in working up the data for a three or four-hour trial becomes excessive.

We may next turn to a brief examination of the changes in condition most likely to occur between the calibration and the

actual trial, and of their probable influence on the results. We will first consider the changes most likely to affect the revolutions, the initial pressure remaining the same. Such are changes in displacement or trim, change in the character of the bottom, change in propeller, either as to pitch, diameter or surface. With all such changes the valve events and expansion ratios remain constant, and the only influence of varying revolutions is by way of the effect due to wiredrawing through the ports and passages as influenced by varying piston speed. Unless the variation in revolutions and piston speed is extreme, it does not seem likely that such influence would be appreciable. So far, therefore, as such changes are concerned, the result would seem to be quite independent of any changes likely to occur.

The next condition to be noted is the quality of the steam. There is no doubt but that the expansion line is somewhat dependent on the amount of water introduced into the cylinder with the steam, or, still otherwise, that under the same pressure and with the same valve events, slightly different cards may result with considerable variation in the quality of the steam. Under ordinary conditions where the quality remains within a range of a few per cent., this influence is not likely to be appreciable, and ordinarily no trouble would be experienced from this cause. Care should be taken however, that on both the calibration and actual trials the regimen of the boilers, the number used, the nature of the feed and all conditions on which quality depends should be as nearly as possible the same. We may also note here the result of sudden and excessive priming. In such case, for the time being, all normal relation between the initial and the mean effective pressures is lost, and the data for such period should be eliminated. This may readily be done, the abnormal period being denoted by the sudden drop in revolutions consequent on this condition. The method is therefore readily applicable to the elimination of such data, and to the determination of a mean free from abnormal influences of this nature.

It therefore appears that the conditions whose constancy is the most important are those which may influence the quality of the steam, and that moderate changes in the ship or propeller are not

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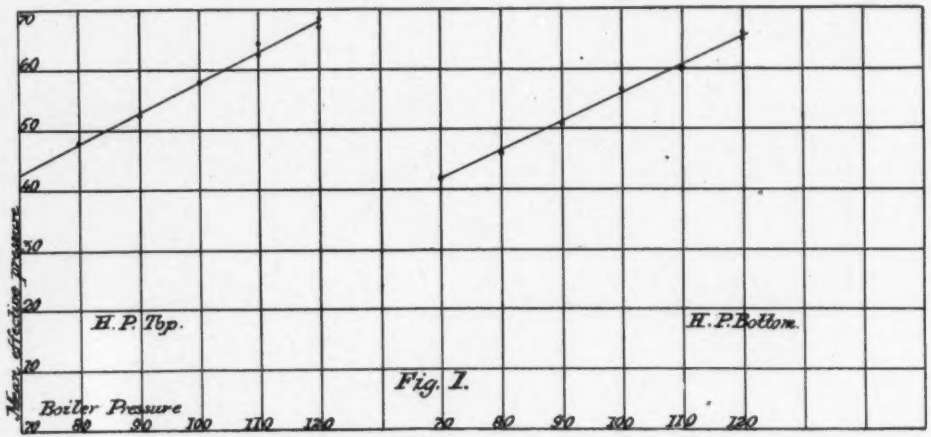

70
60
50
40
30
20
10
Mean effective pressure
H.P. Tbp.
H.P. Bottom.
Boiler Pressure
70 80 90 100 110 120
70 80 90 100 110 120

Fig. I.

20

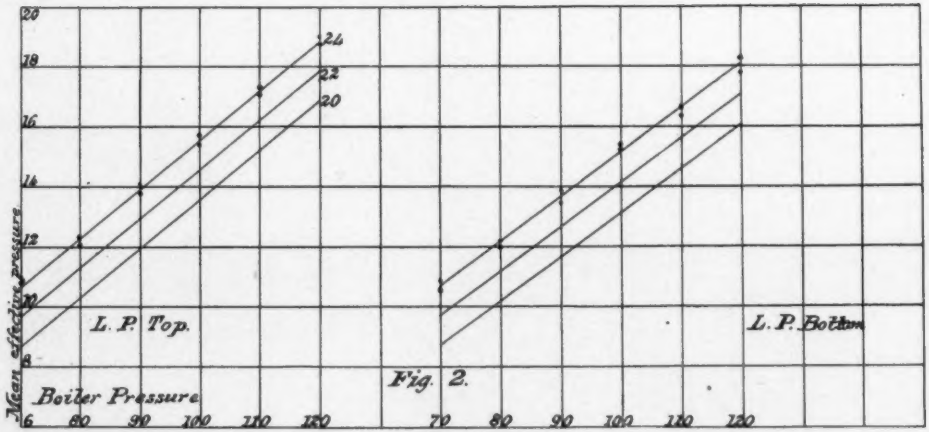
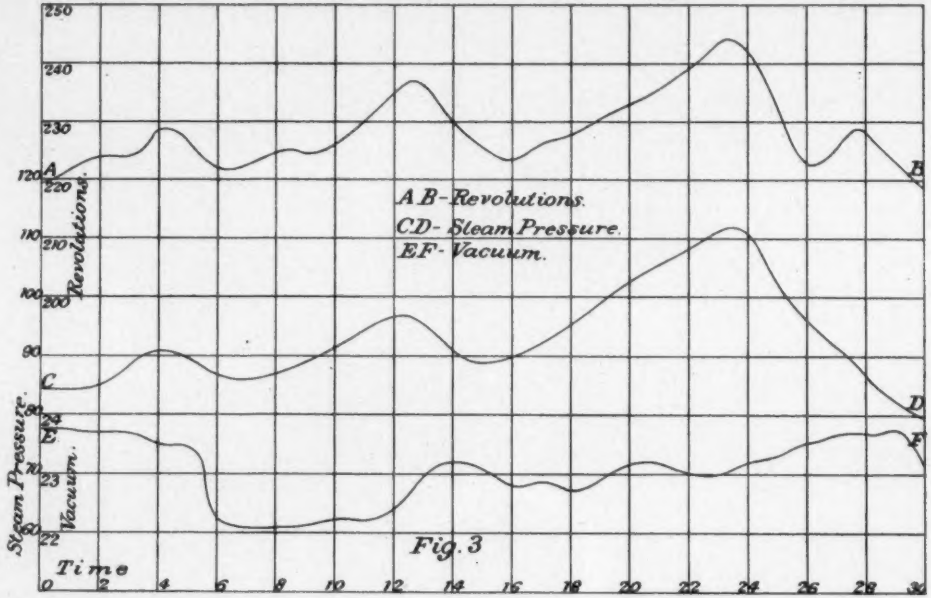

20
18
16
14
12
10
8
6
Mean effective pressure
24
22
20
L. P. Top.
L. P. Bottom
Boiler Pressure
60
80
90
100
110
120
70
80
90
100
110
120

Fig. 2.


250
240
230
120
A
220
B
110
210
Revolutions.
100
200
90
C
80
24
E
70
23
Vacuum.
60
22
Steam Pressure.
A B-Revolutions.
C D- Steam Pressure.
E F- Vacuum.
D
F
Time
0
2
4
6
8
10
12
14
16
18
20
22
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26
28
30

Fig. 3

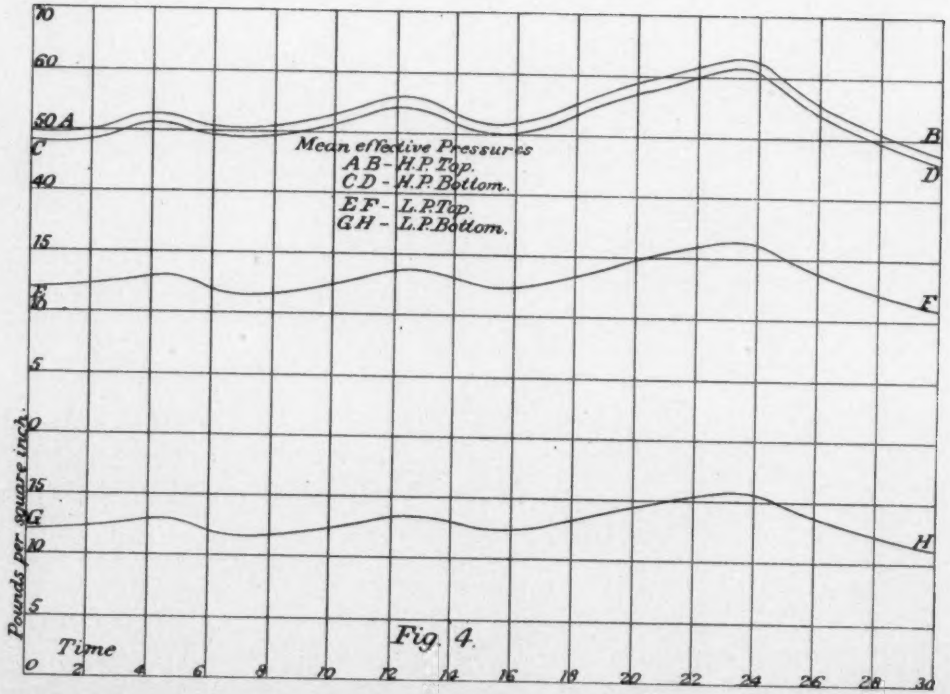

70
60
50 A
C
40
15
E
10
5
0
15
G
10
5
Pounds per square inch.
Time
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
Mean effective Pressures
A B - H.P. Top.
C D - H.P. Bottom.
E F - L.P. Top.
G H - L.P. Bottom.
B
D
F
H

Fig. 4.

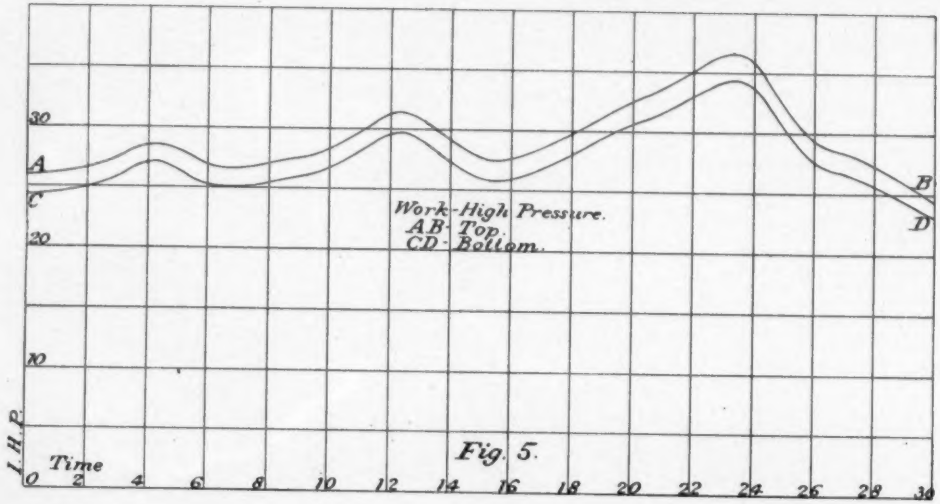
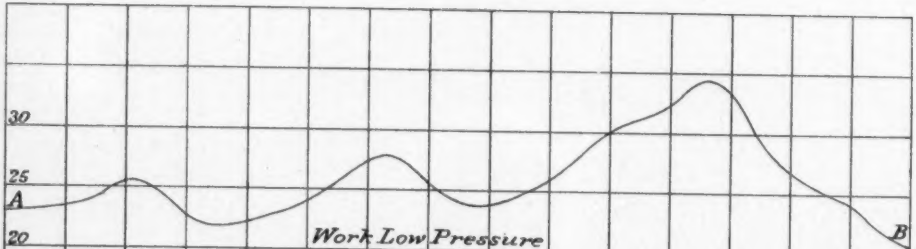

30
A
C
20
10
I.H.P.
Time
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
B
D
Work-High Pressure.
AB-Top.
CD-Bottom.

Fig. 5.


30
25
A
20
Work Low Pressure
B

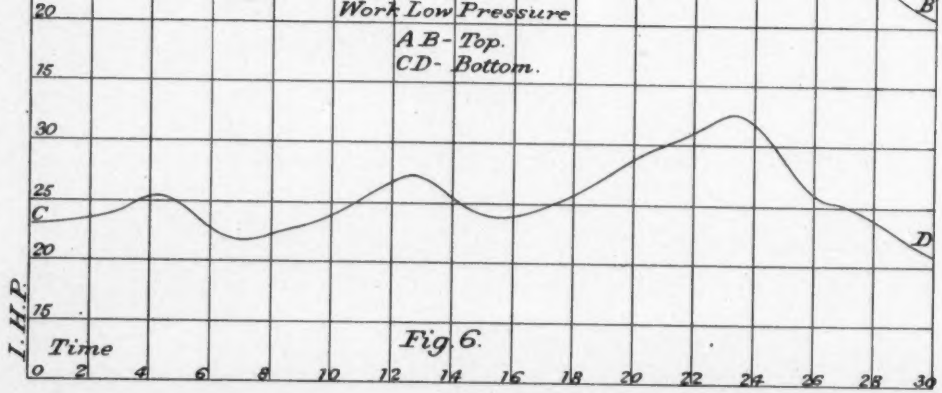
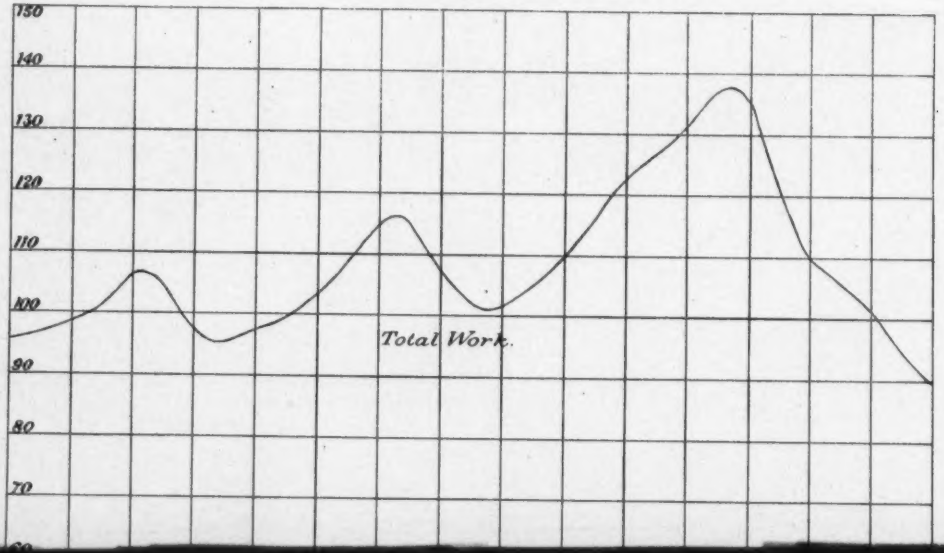

Work Low Pressure
A B- Top.
C D- Bottom.
B
D
C
I. H.P.
Time
20
15
30
25
20
15
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30

Fig. 6.


Total Work.
130
140
130
120
110
100
90
80
70

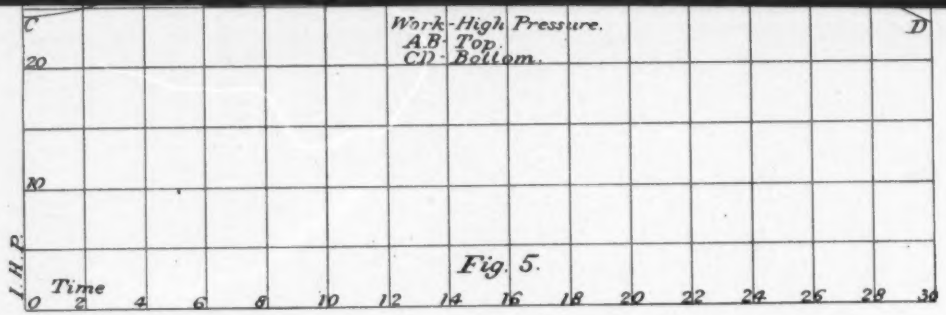
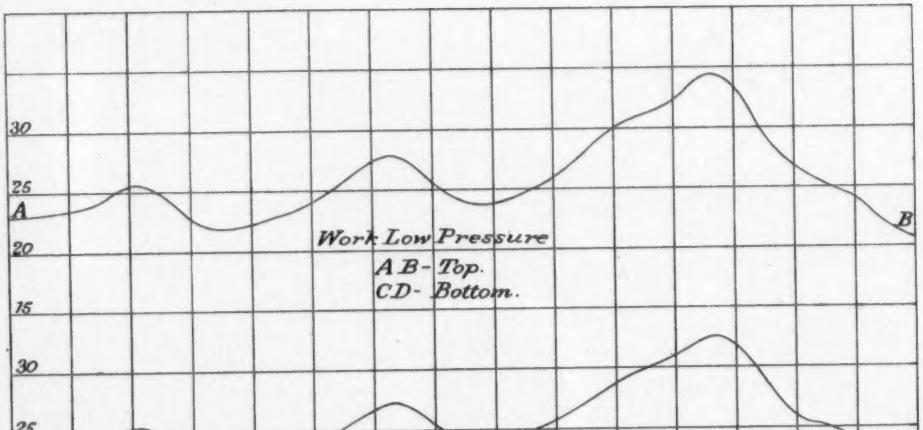

C
D
Work-High Pressure.
A.B- Top.
C.D- Bottom.
20
10
I.H.P.
Time
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30

Fig. 5.


30
25
A
B
Work Low Pressure
A.B- Top.
C.D- Bottom.
20
15
30
25

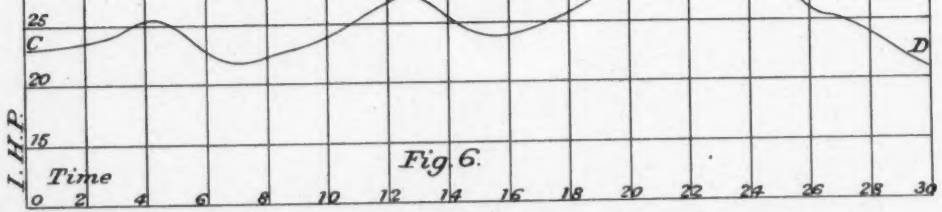

25
C
20
D
15
I.H.P.
Time
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30

Fig. 6.

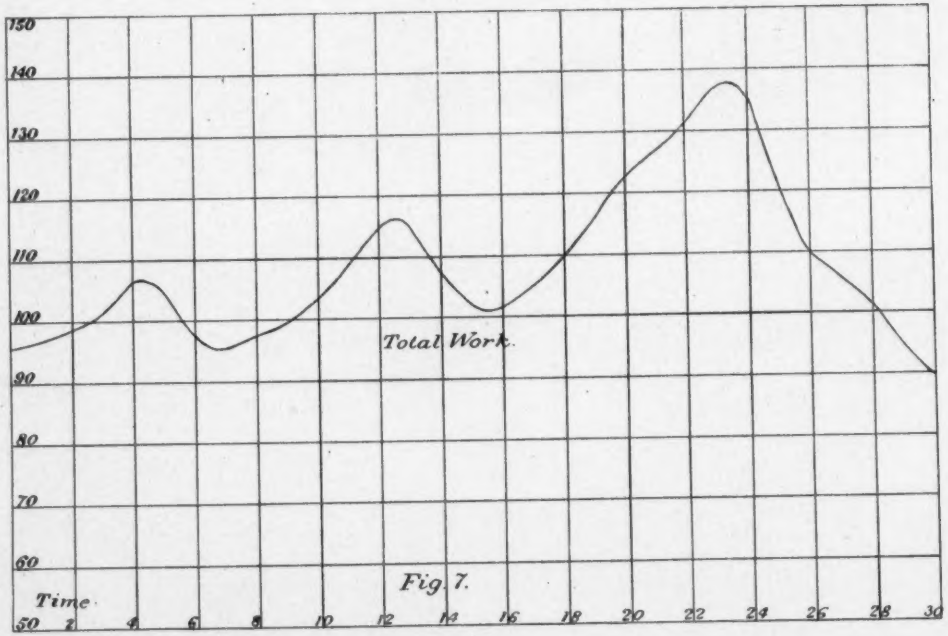

150
140
130
120
110
100
90
80
70
60
50
Total Work.
Time
2 4 6 8 10 12 14 16 18 20 22 24 26 28 30

Fig. 7.

likely to produce changes of any moment in the relations between the initial and mean pressures.

In addition to the uses of this method in conducting a trial, it may be mentioned that the information thus derived might often be of value for the approximate determination of the power from readings taken directly from the steam and vacuum gauges and counter.

So far as the actual true mean value of the power is concerned, the results of these observations indicate, when the firing is fairly uniform and the steam pressure varies slowly through a range of perhaps not greater than 5 to 10 per cent., that we should expect the results as derived in the usual way to give all the accuracy significant in such cases. With wider ranges of variation, such as are shown in the present case, the ordinary method is likely to give errors of considerable amount as discussed above.

As to whether the possibility of this superior accuracy would be of value in any particular case, is a point with which we are not here concerned. It is our purpose to show how a continuous record may be obtained. The question of its use in any given case must be settled by the consideration of the value in such case of the various items of information which such a record might be able to furnish.

THE U. S. S. *TERROR* AND THE PNEUMATIC SYSTEM AS APPLIED TO THE GUNS, TURRETS AND RUDDER.

By Passed Assistant Engineer T. W. Kinkaid, U. S. Navy.

The keel of the United States Monitor *Terror* was laid in 1874 at Cramp's shipyard, Philadelphia. Her sister ships are the *Miantonomoh*, *Amphitrite* and *Monadnock*. The displacement, commonly given as 3,990 tons, is normally about 4,100. The vessel entered upon her first commission on April 15, 1896, at the Navy Yard, New York.

The ship was not given lines for high speed, and eight or nine knots is considered good steaming with clean bottom and smooth water. She can, however, be forced up to ten knots.

GENERAL DATA.

Length on L. W. L., feet and inches	259- 6
Beam, feet and inches	55-10
Draught, maximum, feet and inches	15- 3
mean, feet and inches	14- 6
Tons per inch at L. W. L.	27.67
Thickness of hull plating, inch	$\frac{5}{8}$
of protective deck, inches	$1\frac{1}{4}$
side armor, nickel steel, inches	7
wood backing, inches	20
Number of bulkhead water tight compartments	18
double bottom compartments	31
Mean free board, inches	27

The engines are of the inclined, compound, direct-acting type, completely jacketed. The starboard cylinders belong to the port shaft, and *vice versa*. Each L.P. piston has two rods, $4\frac{1}{8}$ inches diameter, and each H.P. piston has one rod, 5 inches diameter. None of the bearings contain antifriction metal. The condenser is underneath of and forms the bed for the two main engines. Each cylinder is fitted with Meyer cut off valves. The rods are packed with Katzenstein's metallic packing.

MACHINERY DATA.

Diameter, H.P. cylinder, inches	32
L.P. cylinder, inches	48
Stroke, inches	42
Length of connecting rod, inches	84
Cooling surface in condenser, square feet	4,900
Diameter and stroke of circulating pump cylinder, inches	11 and 12
of runner for circulating pump, inches	35
of nozzle, inches	14
and stroke of two air pump steam cylinders, inches	11½ and 24
and stroke of two air pump water cylinders, inches	16 and 24
Number of single-ended, 3-furnace boilers	6
Total grate surface, square feet	378
heating surface, square feet	8,781
Diameter of shell, feet	12
Length of shell, feet and inches	10– 7
Thickness of shell, inch	¾
Working pressure, per gauge, pounds per square inch	80
Diameter of smoke pipe, feet and inches	8– 2
Height of smoke pipe above grate, feet and inches	52–11
Number of fire room blowers	4
Diameter of fans, inches	50
and stroke of steam cylinders (2), inches	5 and 5
Area of outlet, square feet	4

There is no evaporator.

The bunker capacity is nominally 285 tons of bituminous coal; but, on account of the presence of numerous ventilating ducts, 265 tons is the actual maximum.

VENTILATING SYSTEM.

The ventilating system is elaborate. Two of the 50-inch fans which supply the fire room are ordinarily used to exhaust from the forward berth deck. The other two fire room fans take fresh air from the armored duct and deliver part to the fire room and the remainder for general ventilation; or, all of the fresh air may be directed to either use alone. Two more 50-inch fans, located in the engine room, exhaust from the after berth deck and deliver into the enclosed spaces back of and over the boilers, whence air is discharged through holes in the armored deck into a casing surrounding the smoke pipe. All of the six main ventilating blowers can be arranged to draw fresh air from above. The dis-

charge of air into the spaces above the boilers is intended to ameliorate the temperature of the superstructures. The latter contains the crew's mess room, galley, water closets, pay office, armory, sick bay, executive officer's office, captain's office, warrant officers' mess room, and one state room.

Two 30-inch steam fans, one for exhaust and the other for supply, are provided in the engine room for the special ventilation of that compartment.

The bunkers are supplied with fresh air through pipes which tap the downcasts to blowers. The exhaust from the bunkers is led to a breeching at the bottom of a 2-foot pipe concentric with the smoke pipe. This draft pipe also serves to draw some of the hot air from the fire room. A cone valve at the top is worked by a rod and lever in the fire room.

A 24-inch supply main runs fore and aft the ship on either side and admits air to the berth deck through louvres in the deck. Branches lead to the various store rooms and magazines.

Parallel to the supply mains run the exhaust mains, with branches overhead on the berth deck. All the living spaces on the berth deck have both supply and exhaust ventilation.

In addition to what has been already mentioned, the *Terror* has the usual auxiliary machinery.

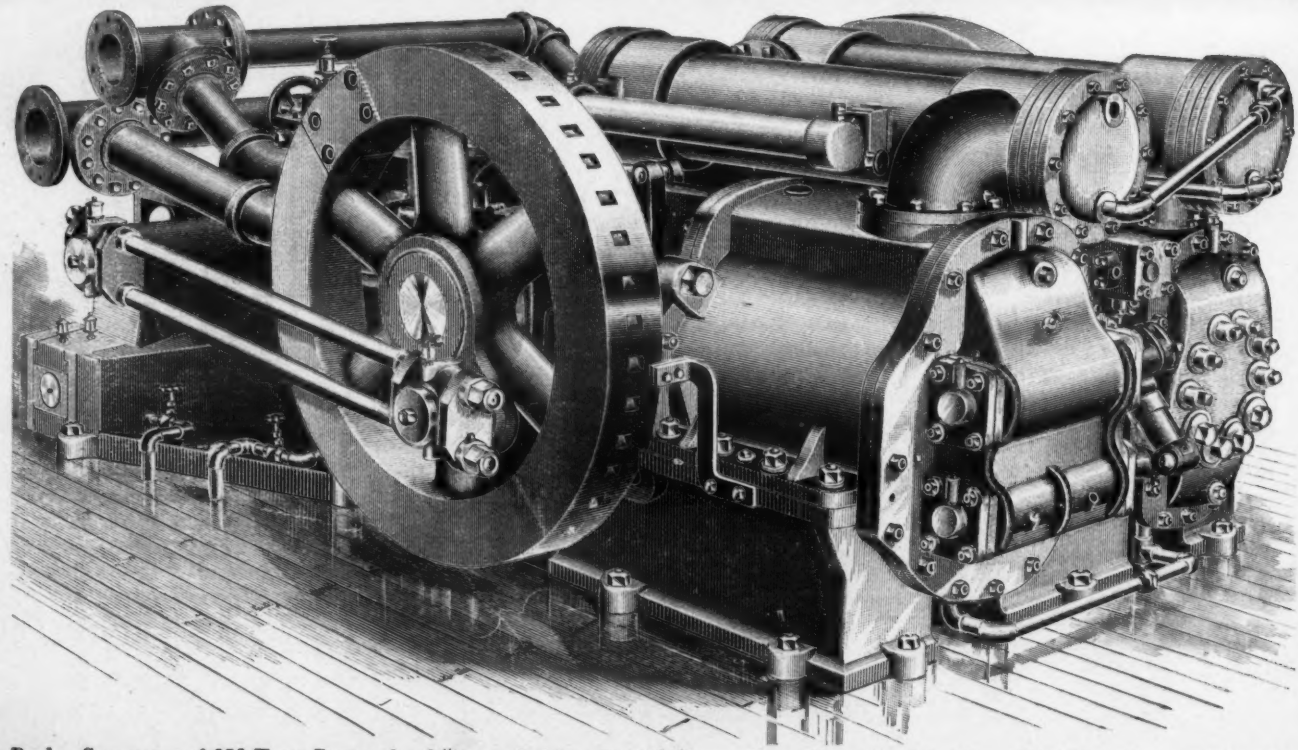
ARMORED STRUCTURES.

The armored structures on the *Terror* consist of two main turrets, armored smoke pipe and armored ventilating tube.

Diameter of each turret, external, feet and inches	24–2
Thickness of turret armor, nickel steel, inches	11½
Height of turret armor, feet and inches	6–3
Thickness of smoke pipe and ventilator armor, inches	6
Height of smoke pipe and ventilator armor, feet	4
Internal diameter of armored ventilator, feet and inches	5–6
Weight of one 10-inch gun, pounds	56,400
turret and contents, tons, about	250

PNEUMATIC SYSTEM.

Of course, the foundation of a good pneumatic plant is a proper contrivance for furnishing an abundant supply of compressed air.



Duplex Compressor of 250 Horse Power, placed "'tween deck" on U. S. Monitors. Length over all, 14 ft. 4 ins. Width 8 ft. 8 ins. Height, 5 ft.

The two compressors on the *Terror* are special designs of the Norwalk Iron Works, and are shown in the cut. They were put into the ship about three years ago. Necessarily, they are more compact than the ordinary commercial form of machine. They are very heavy and there is considerable friction; but the machines are not too large for extreme requirements.

Number of compressors	2
Diameter of steam cylinders (2), inches	21
H.P. air cylinder, inches	17½
L.P. air cylinder, inches	28
Special air plunger, inches	2⅝
Stroke of all pistons and plunger, inches	24
Floor space occupied by one compressor, feet and inches	14-6 × 9-6
Maximum height of compressors, feet and inches	5-6
Weight of one compressor (estimated), pounds	24,000

Although each turret has one of the compressors near by, a communicating pipe between the machines allows either compressor to operate either or both turrets. The receiver pressure in the compound compressor is about 26 pounds per gauge, and the final pressure, used for the ordinary purposes of the system, is 125 pounds per square inch. The small plunger operated by the machine takes air at 125 pounds, and compresses it to 1,200 pounds or more, as may be required; but 550 pounds is about the right pressure for the recoil cylinders. The mean recoil with this pressure has been about 30 inches. The high pressure service is needed only in case of leakage from the recoil cylinders. The intake valves on the L.P. air cylinder are simple cast iron Corliss valves, all the other valves on the compressor being spring-seated mushroom valves. The air cylinders and the two receiver cylinders are water-cooled, a small duplex pump, with steam cylinder 4½ inches, water cylinder 3½ inches in diameter, and stroke 4 inches, giving the necessary circulation for both compressors. The communicating pipe, after a short service, becomes too hot to be borne by hand.

The greatest tax on the compressors is caused by the turning engines, of which there are two in each turret. The motors run at high speed and are reversed and stopped by means of the well-known shifting eccentric, operated by levers from the raised plat-

form between the guns. The motor shaft carries a worm which gears into a bronze wheel held by friction to a vertical shaft. The latter is spur-geared to a second shaft carrying the pinion which operates the turret by contact with a large internal cast steel gear. Should the turret receive a heavy turning impulse, such as would be imparted by a glancing shot, for instance, it is expected that the friction connection of the bronze worm wheel would save the gear teeth from rupture. The weight of the turret is borne on conical bronze rollers carried in a circular cage.

The storage capacity of the pipe leading from either compressor to its neighboring turret is so small that the turret can be moved within two or three seconds of the time of starting the compressor. When the compressors, on the recent sea trial, were driven to their utmost capacity, in order to establish a record for turning the turrets, their speed at first was about 96 revolutions per minute, but this was somewhat reduced by priming.

Two automatic stops, secured to the floor of each turret chamber, operate on the reverse shaft of the training motors and prevent the turret from turning too far.

Diameter of communicating main, inches	8
Storage capacity of communicating main, cubic feet, about	60
Diameter of recoil service main, nominal, inches	2
L.P. service pipe to each turret, inches	2½
recoil service pipe to each turret, inches	1½
cold water pipe to each compressor, inches	1½
Number of training motors to each turret	2
air cylinders, each motor	2
Diameter and stroke, inches	7 and 8
Ratio of gearing	1 to 1,142.4
Pitch diameter of internal gear of turret, feet and inches	14–10
pinion, inches	14 1/16
Number of teeth of pinion	10
Diameter of pinion shaft, inches	6
Time of train through 270 degrees, least, seconds	52
slowest, minutes	21
Least air pressure required to operate motors, pounds per square inch	20

The elevation of the gun is controlled by a ram with rounded head bearing on the lower side of the rear transom of the gun carriage. This ram is operated by glycerine (80 per cent. glyc-

erine, 20 per cent. water), fed from a vertical tank inside the turret. A glass gauge near the top of the tank shows the level of the glycerine within; and the requisite working pressure is obtained from the air admitted through a small pipe in the top of the tank. The admission and exhaust valves are balanced slide valves operated by suitable levers. The valves are situated both in the air and the glycerine pipes, those in the latter giving the desired lock action.

The exhaust for the glycerine is back into the tank.

Diameter of elevating ram, inches	18
piping, glycerine, inches	2
air, inch	½
Shortest time from greatest elevation to greatest depression, seconds	32
depression to greatest elevation, seconds	17
Preponderance of gun, pounds	25,000

It will be seen that a sufficiently slow motion can be given to the ram to secure a fine adjustment of elevation.

The ammunition car travels on curved Z-bar guides provided with notches which engage with pawls on the car. The pawls are held against the notches by springs during the ascent, thus preventing the accidental fall of the car. The gun carriage supports a heavy bracket which not only carries the rammer, but provides the necessary stop for the car, so that the car may be always held with its lower chamber in line with the rammer and the bore of the gun. The rams for the ammunition cars are located between the two guns. They carry six multiplying sheaves. The hoisting rope reeves over a sheave co-axial with the horizontal pivot of the gun carriage; consequently the position of the car relative to the bore and the rammer does not change during a change of elevation of the gun.

The three chambers of the ammunition car are held in a ring, mounted on rollers and sustained in one of three positions by a spring catch. When the car first ascends, the lower chamber carries the projectile, the other two chambers containing the two powder cartridges. The stop on the bracket maintains the car in such position that the lower chamber is in line with the bore of the gun. A stroke of the rammer sends in the projectile, a pe-

culiar thud indicating that the shot or shell is home. The catch is released and one of the cartridge chambers is revolved, almost by gravity, into the loading position. After the full charge is in the gun and the car lowered, a turn of a valve slides the breech plug into line and the rammer sends it in. The necessary sixth of a turn to lock the plug is given by hand.

As soon as the gun is fired, it is returned to battery by the compressed air in the recoil cylinders. The rammer reaches out, a rotary catch on the plug is moved, and the return stroke of the rammer withdraws the plug. The plug being withdrawn, the catch does not interfere with the transverse motion of the plug motor, which now moves the tray and plug to one side. A jet of air commences to play into the bore to drive out the smoke.

Down in the loading room the shell is run out on a trolley to meet a loading tray swung on the central column. When the tray has been rotated sufficiently, the movement of a vertical ram, by means of rack and pinion motion, elevates the tray and cants it sufficiently to cause the projectile to slide down into its proper chamber in the loading car. The two powder charges of 120 pounds each are easily put in, and the car is ready to rise. Just before the car completes its ascent, it strikes and throws into the lower slot of the screw box a hinged filling piece, called a loading tray, which gives a flush surface on which to push in the projectile and powder. After the last bag of powder is rammed in, the filling piece is automatically swung back out of the way by the action of a cam on the breech plug motor as it slides the plug over into position. The telescopic rammer is constructed of composition. Future designs will call for some harder material, probably tool steel, which will withstand upsetting action. The withdrawal of the rammer is accomplished by the proper location of small holes through each tubular element, which, while the main portion of the telescope cylinder is exhausting, allow air to enter successively the spaces formed between contiguous tubes and bounded by their collars. The location of these holes is chosen to ensure a proper cushion. When the rammer is extending, it is impossible to tell which tube will protrude first; but, in closing up, the larger tubes must come in first.

The breech plug motor is supplemented by a hand screw, which can be thrown out of action by lowering a half nut attached to the motor frames. The motor is somewhat similar in principle to one element of the telescopic rammer.

Diameter of ammunition hoist cylinder, inches	15
Stroke, inches	34¾
Diameter of sheaves, inches	19
Number of sheaves	6
Multiplication of stroke	6
Diameter of steel rope, 6 strands, inch	¾
Time of raising loaded car, seconds	45
Diameter of pipe to hoist cylinder, inch	½
largest tube of rammer, inches	9¼
smallest tube of rammer, inches	2
Greatest extension of rammer, inches	126 11/16
Diameter of air pipe to rammer, inch	¾
holes for withdrawal stroke, inch	⅜
jet for bore, inch	⅛
breech plug piston, inches	3½
Time of withdrawing breech plug and swinging it to one side, seconds	17

The recoil cylinders, two to each gun, are secured to the gun saddle. The pistons within the cylinders remain stationary; the cylinders recoil with the gun. The piston rods, which are secured to the rear transom of the carriage are large and stiff, so large in fact that, with the air pressure the same on both sides of the piston, the gun is kept run out. When the gun is fired, there is, of course, a banking up of air pressure before the piston, but an ingenious valve rod, tapered in the middle, advances with the cylinder and allows sufficient air to pass from one side of the piston to the other to restrain the rise of pressure within the desired limits. The valve rod is full at each end, so that a proper cushion is ensured at the limits of the recoil and of the counter recoil. The exact proportions of the valve rod were determined experimentally. The action of the recoil mechanism is very satisfactory and has been much admired.

A check valve on the supply port at the end of the recoil cylinder prevents undue leakage, and the recoil cylinders have stood for weeks with an almost imperceptible loss of pressure.

The air for the rammer and plug motor is carried to the gun

carriage, near its pivoted end, in pipes with hose connection, but the recoil service pipe is rendered flexible by being coiled like a spiral spring, and thus hinge joints are dispensed with. A telescope joint is used for the breech plug motor supply pipe.

Diameter of recoil piston, inches	14
piston rod, inches	6
valve rod, greatest, inches	3
Recoil with full charge, about, inches	30
Weight of powder charge, pounds	240
shell, pounds	500
Muzzle velocity, feet per second	2,000
energy, foot tons, about	14,000
Energy of recoil, estimated from indicator diagram and friction allowance, foot pounds	650,000
Mean velocity of recoil, feet per second	24
counter recoil, feet per second	8
Maximum pressure in recoil cylinder, pounds per square inch	1,300
Mean effective pressure, pounds per square inch	800
Pressure required to run gun out on level, pounds per square inch	250
Pressure in recoil cylinders, when, at 13 degrees elevation, the gun comes in by gravity, pounds per square inch	70
Pressure required to run gun out at 13 degrees elevation, pounds per square inch	570

The central column in each turret carries sleeves which receive the air from the supply pipes and distribute it to the various motors above. The center of the column carries a shaft for operating the steering mechanism. Each turret also contains a lever and rheostat for electric steering; the wires being led down the central column.

There is not much head room on the upper turret platforms; and the forward turret, in the absence of a conning tower, must accommodate not only the turret officer but the captain as well.

There is at present no direct signal gear to the engine rooms, a voice tube relay through central station being depended upon.

As the maximum train of the turrets is only 270°, there would be no difficulty connected with the lead of additional electric wires and flexible voice tubes.

The pneumatic steering gear is simple and powerful. Two athwarthship cylinders, whose pistons are attached to the same heavy piston rod, are firmly secured to castings built into the

ship; the round stern of the *Terror* being peculiarly favorable to the athwartship arrangement. At midlength, the piston rod carries a slotted head, the brasses of which permit the sliding of the tiller as its angularity changes. The two cylinders are respectively to starboard and to port, and only their outboard ends are utilized as motors. The common piston rod is so large that only a small area is left on the inboard face of each piston. The inboard ends of the cylinders are connected by a pipe in which there is a controlling piston valve, operated by the same mechanism that admits air to and exhausts it from the outboard ends of the cylinders. The distribution of air is effected by a simple slide valve receiving its motion from a "float lever," one end of which is controlled by an arm projecting from the rudder head. The float lever receives its initial motion from a threaded spindle working in a nut on the lever. The spindle can be revolved in various ways, to be mentioned hereafter. When the distributing slide valve is in mid position so also is the controlling valve; the latter preventing any exchange of air between the inboard ends of the cylinders. Any rotation of the threaded spindle moves both the distributing valve and the controlling valve, one valve supplying the power and the other relieving the elastic lock.

The threaded spindle, which operates the float lever, lies in a fore-and-aft direction and carries various clutches, all but one of which engage with sprocket wheels. At the end of the spindle is a clutch which engages with an electric motor geared down by worm gearing. The wires from the motor lead to the rheostat at the steering stations in each turret and in the pilot house. The same stations are provided with small wheels which give motion to small wire ropes led to drums in the steering room. Each drum has its separate sprocket chain connection to the screw spindle of the steering gear. So long as there is a supply of air, the principle of the float lever insures the motion of the tiller in obedience to that of the screw spindle.

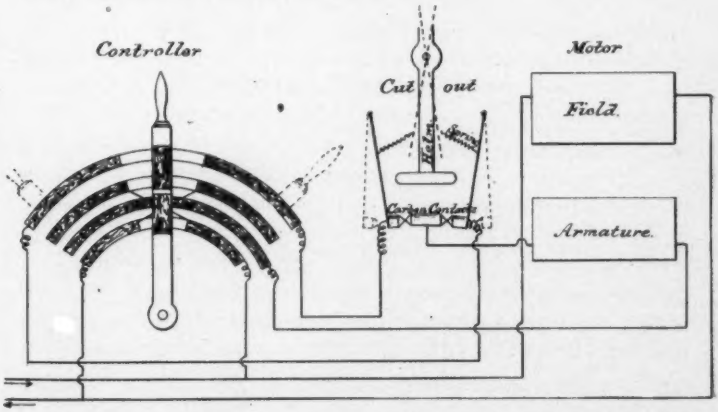
Each steering wheel is provided with a geared index which acts as a tell tale or indicator of the rudder's motion, or, rather, of the motion which the rudder should have as a result of the motion imparted to the screw spindle of the float lever. It is proposed

to connect the tell tales directly to the rudder post. The officer of each gun has under his feet, just above the central column, a circular index with two concentric circles, one showing rotation of turret, the other the helm angle.

In order to deaden the noise of exhaust from the various pneumatic cylinders, the exhaust pipes have been arranged to discharge into vessels or pipes containing pebbles, a simple muffler which is very efficient.

Diameter of steering pistons, inches	14
Stroke, hard over to hard over, feet and inches	5–10
Lap of slide valve, inches	0
Exhaust lap, inch	$\frac{1}{32}$
Least tiller radius, feet	4
Least time, hard over to hard over, 120 pounds air pressure, at dock, seconds	4
Area of rudder, square feet	81.2
Width of rudder, unbalanced, feet and inches	7– 9
Diameter of piping, inches	$1\frac{1}{2}$
Minimum air pressure that has been used for steering in smooth sea, pounds per square inch	30
Diameter of drum ropes, inch	$\frac{3}{4}$
Least pressure to move steering gear at dock, pounds per square inch	5

All sheaves have ball bearings. There is a carbon contact, electrical cut-out operated by an arm on rudder head. This keeps the steering motor from overruning.



Reversing Rheostat and Cut-out for Steering Gear Motor.

The packing for nearly all face joints is $\frac{1}{16}$-inch "common sense" sheet metal. On the steam cylinder heads, copper gaskets are used; while on the air heads, paper is employed. For most of the rams and pistons, Tuck's square packing or machine-braided hemp is used. The pump plunger for the high pressure service is packed with cup leathers, lubricated by a pipe from an oil reservoir. The breech plug motor is also leather-packed.

For the low pressure service (125 pounds), single thickness, iron steam pipe is used, except in some places where extra strong pipe is needed to provide against external wear and tear. For the high pressure service (550 pounds), extra strong and double extra strong pipes are used, according to location.

The pipes in the turrets are all of very moderate diameters and are not in any way obstructive.

DATA OF PERFORMANCE.

Date of experimental firing, after turret		August 28, 1896
1st Shot.	Elevation of gun, 8 degrees. Charge, 75 pounds brown prismatic powder.	
	Initial velocity, foot-seconds	817
	Pressure in recoil cylinders, pounds per square inch	350
	Recoil, inches	11⅜
2d Shot.	Same elevation. Charge, 120 pounds.	
	Initial velocity, foot-seconds	1,210
	Pressure in recoil cylinders, pounds per square inch	350
	Recoil, inches	22½
3d Shot.	Same elevation. Charge, 178 pounds.	
	Initial velocity, foot-seconds	1,650
	Pressure in recoil cylinders, pounds per square inch	425
	Recoil, inches	30
4th Shot.	Same elevation. Charge, 240 pounds.	
	Initial velocity, foot-seconds	2,000
	Pressure in recoil cylinders, pounds per square inch	550
	Recoil, inches	30½
Date of experimental firing, forward turret		August 31, 1896
1st Shot.	Elevation, 8 degrees. Charge, 120 pounds.	
	Pressure in recoil cylinders, pounds per square inch	350
	Maximum pressure in recoil cylinders, pounds per square inch	525
	Recoil, inches	21¾
2d Shot.	Same elevation. Charge, 178 pounds.	
	Pressure in recoil cylinders, pounds per square inch	425
	Maximum pressure in recoil cylinders, pounds per square inch	800
	Recoil, inches	33

3d Shot.	Same elevation. Charge, 240 pounds.	
	Pressure in recoil cylinders, pounds per square inch	550
	Maximum pressure in recoil cylinders, pounds per square inch	800
	Recoil, inches	32½
4th Shot.	Charges, 240 pounds each. Both guns simultaneously.	
	Pressure in recoil cylinders, pounds per square inch	550
	Elevation, degrees	12
	Recoil, inches	29 and 32$\frac{9}{16}$

Times of loading, Nov. 21, 1896. Best time from fire to fire, 1 minute and 37 seconds. Times of three shots from left gun, after turret, 1st shot, 1 minute and 55 seconds; 2d shot, 1 minute and 48 seconds; 3d shot, 1 minute and 49 seconds.

The pneumatic system for working the guns, turrets and steering gear is believed to possess advantages over the hydraulic and electric systems. Some of these advantages may be itemized as follows:

1. There is absolute immunity from freezing; which advantage must, of course, be shared with electric systems.

2. The operation of the various motors contributes to the comfort of the men, as the exhaust improves the local ventilation and gives a reduction of temperature which is usually desirable. Hence the physique of the men is not impaired in time of action.

3. Leakage is readily detected and stopped. Should it continue, there would be no distressing increase of temperature, as with steam; or burning out of fuzes, as with electricity; or drenching of the person and compartment, as with the hydraulic system.

4. There is no danger from burns or electric shock.

5. Motors can be started up instantly and stopped instantly without much danger of breakage. In this respect there is an advantage over steam and hydraulic systems.

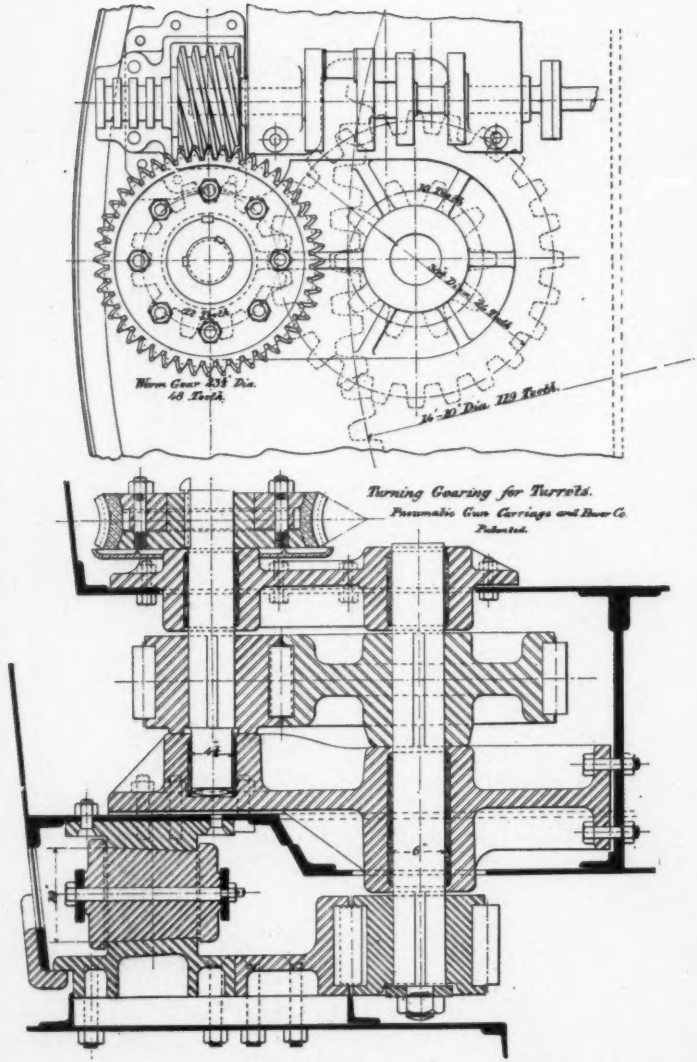
Although the installation of the *Terror's* pneumatic system has extended over a period of years, the original contract calling for the completion of the work in 1889, yet much of the delay has been caused by changes in design, both on the part of the Government and of the contractors. The experience already gained with the present plant ensures the possibility of rapid fulfillment of any future contract that may be made, and it is certain that a future design will show much superiority over that of the *Terror's*

outfit. The system as now in use has been subjected to many alterations, some of them quite radical, and at this writing, the revolving loading car and breech plug motor are undergoing improvement. A non-revolving loading car, which promises to afford greater convenience and speed of loading, has already been designed. A small independent air compressor is being put up in the engine room to supply the air for steering.

The time records given in this paper do not represent the best possibilities of the present installation, as the men had had but a few days' acquaintance with the completed machinery when the record firings were made.

The system is the invention of Mr. H. A. Spiller, vice president and engineer of the Pneumatic Gun and Power Co., of which Mr. W. R. Creecy is president. The work of installation has been under the personal supervision of Mr. E. D. Eldridge, who has had much experience with this line of work under former Government contracts.

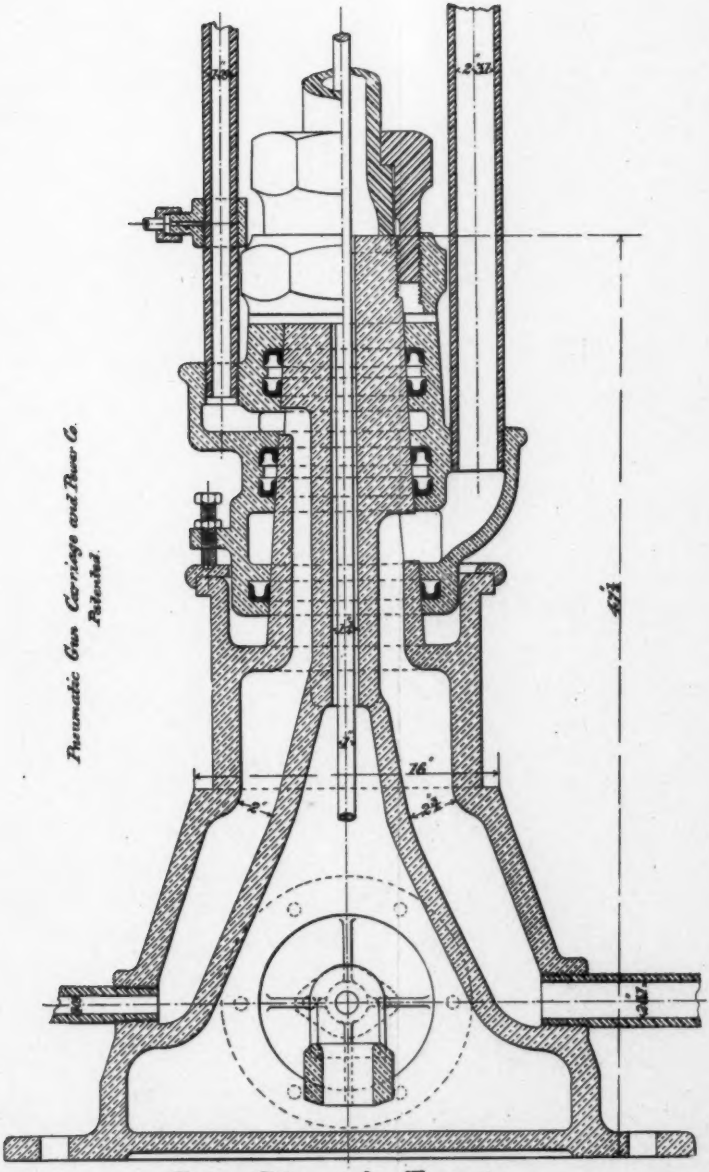
Many naval officers who have witnessed the performance of the *Terror's* pneumatic system would doubtless be willing to adopt the system for use in ships now building, without giving any special consideration to the question of the weights involved. But Mr. Spiller is of the opinion that a new design, involving the latest pattern of high speed compressor and possibly reheating of the air at the training motors, could effect a saving in weight of at least ten per cent. over any hydraulic system now in use. It is obvious that a pipe of given size can transmit a greater power when used as an air conductor than when it is used, with the same factor of safety, for an hydraulic system. In addition, there are the two important facts that, with air transmission, no return pipe and no supply tank are necessary.



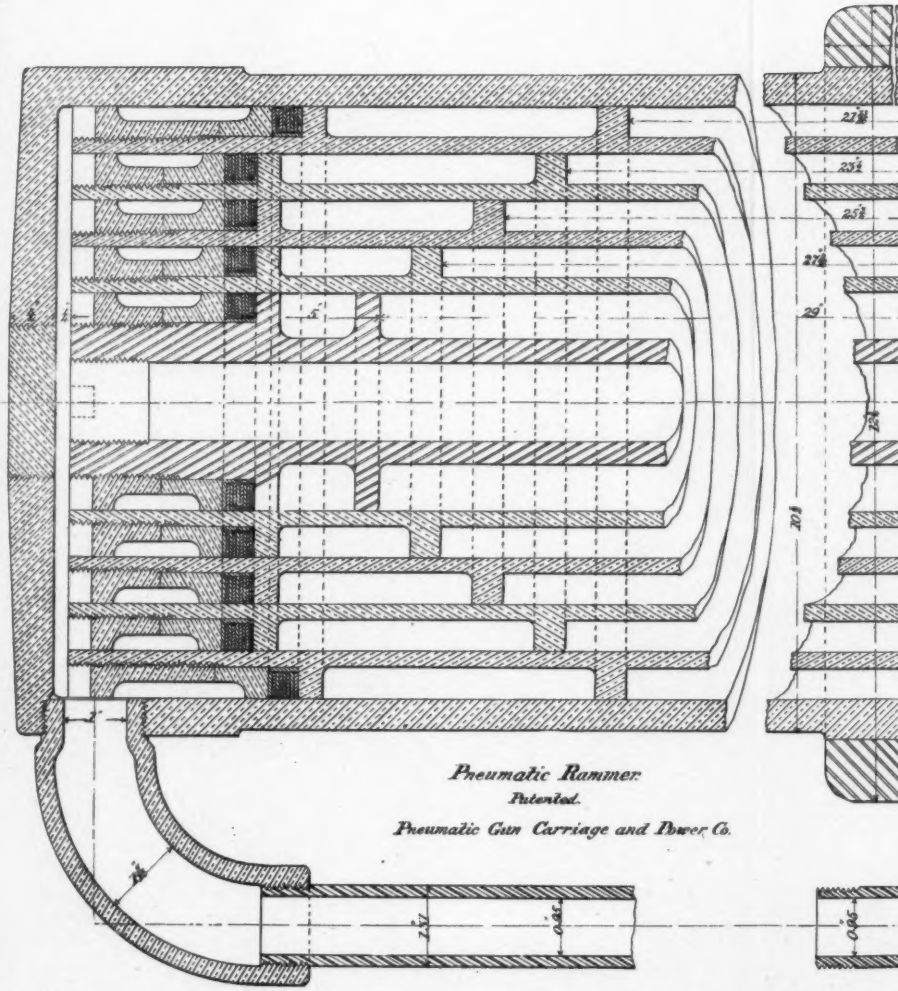
Turning Gearing for Turrets.

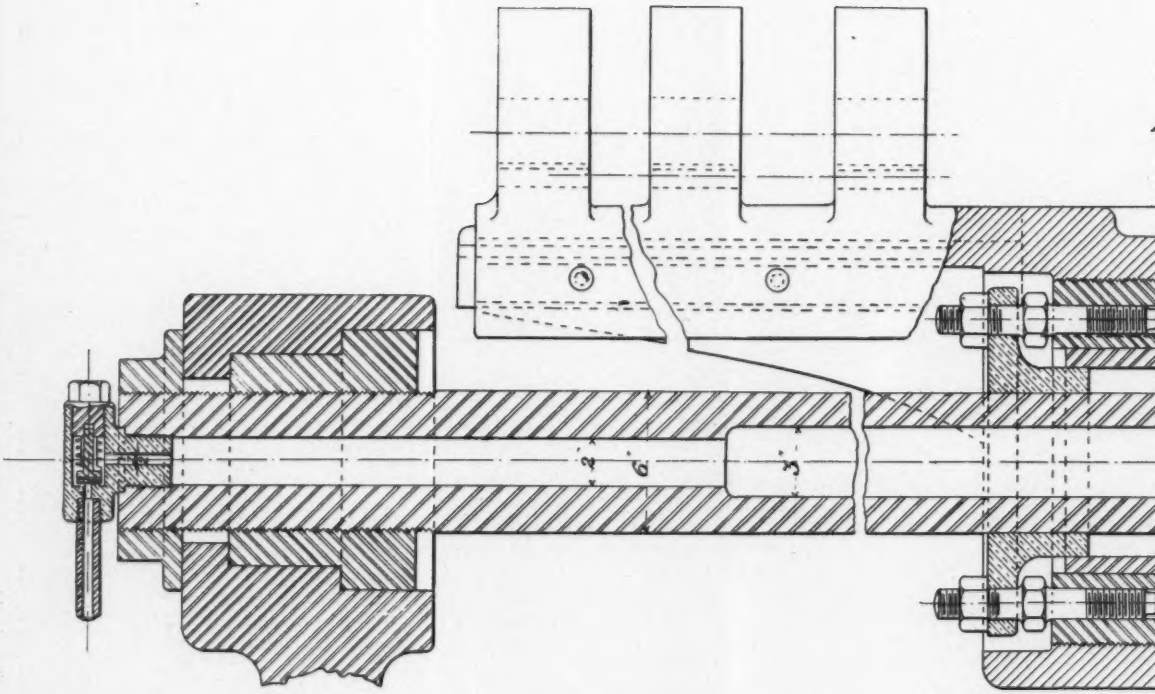
Pneumatic Gun Carriage and Power Co.

Patented.



Center Column for Turret.

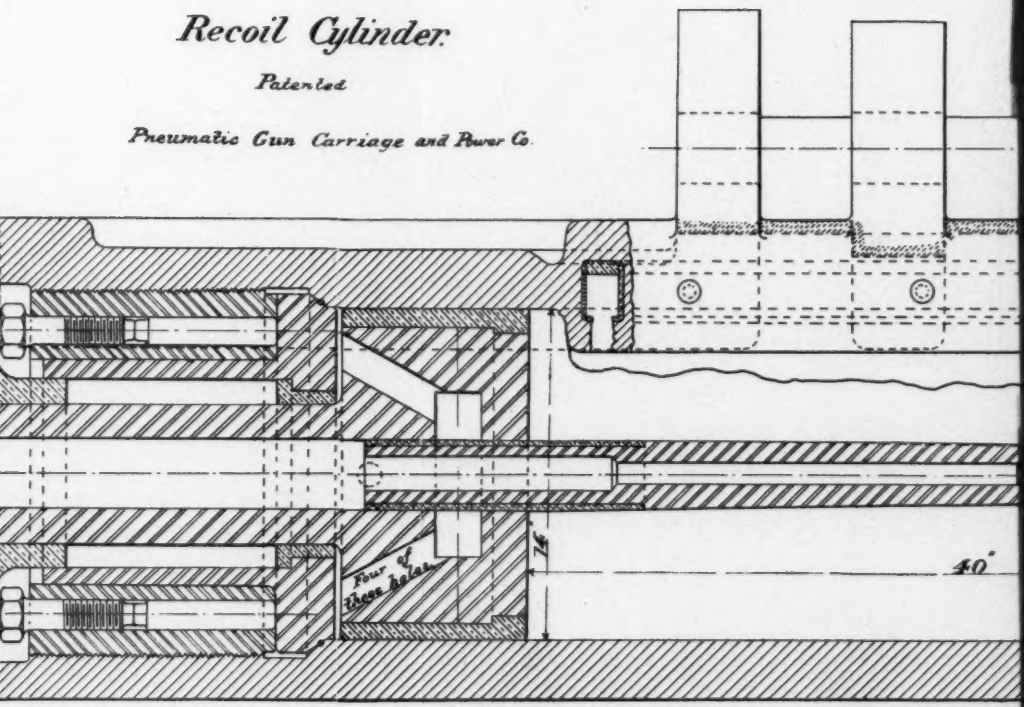

Pneumatic Rammer.
Patented.
Pneumatic Gun Carriage and Power Co.



Recoil Cylinder.

Patented

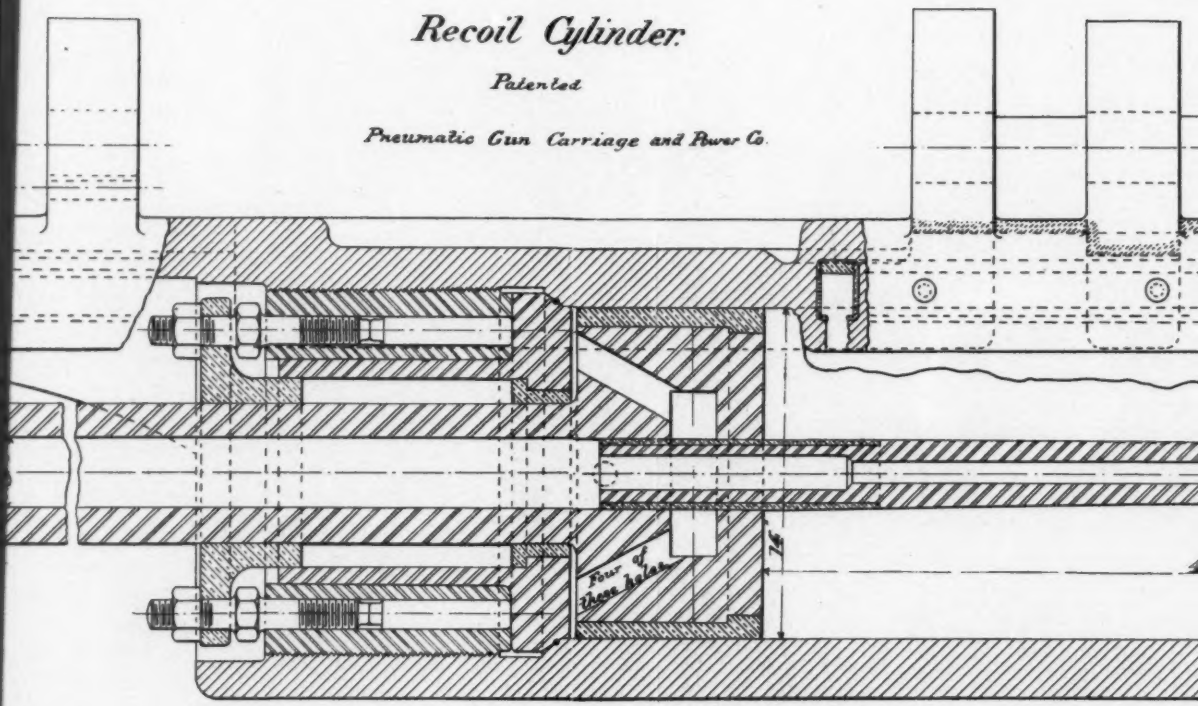
Pneumatic Gun Carriage and Power Co.

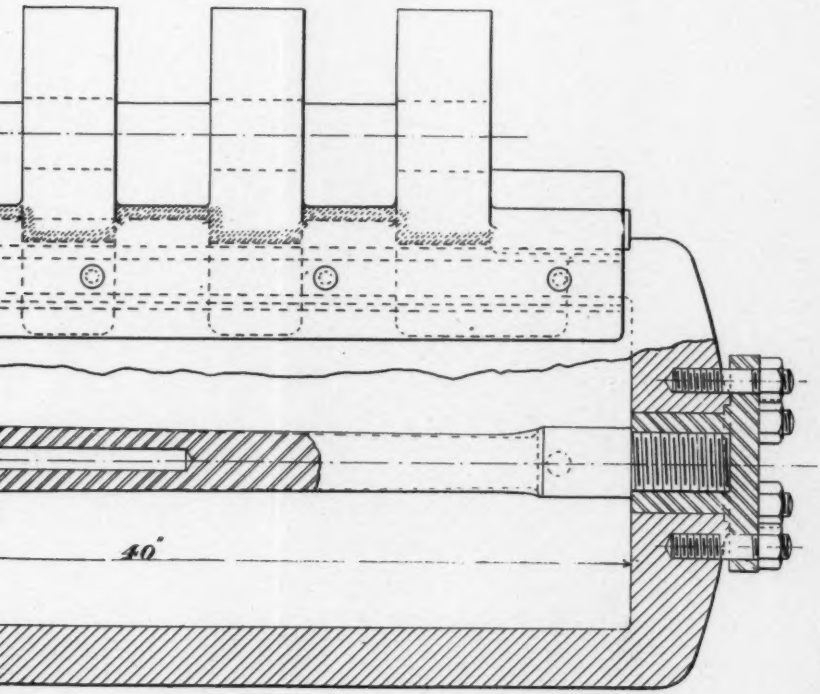


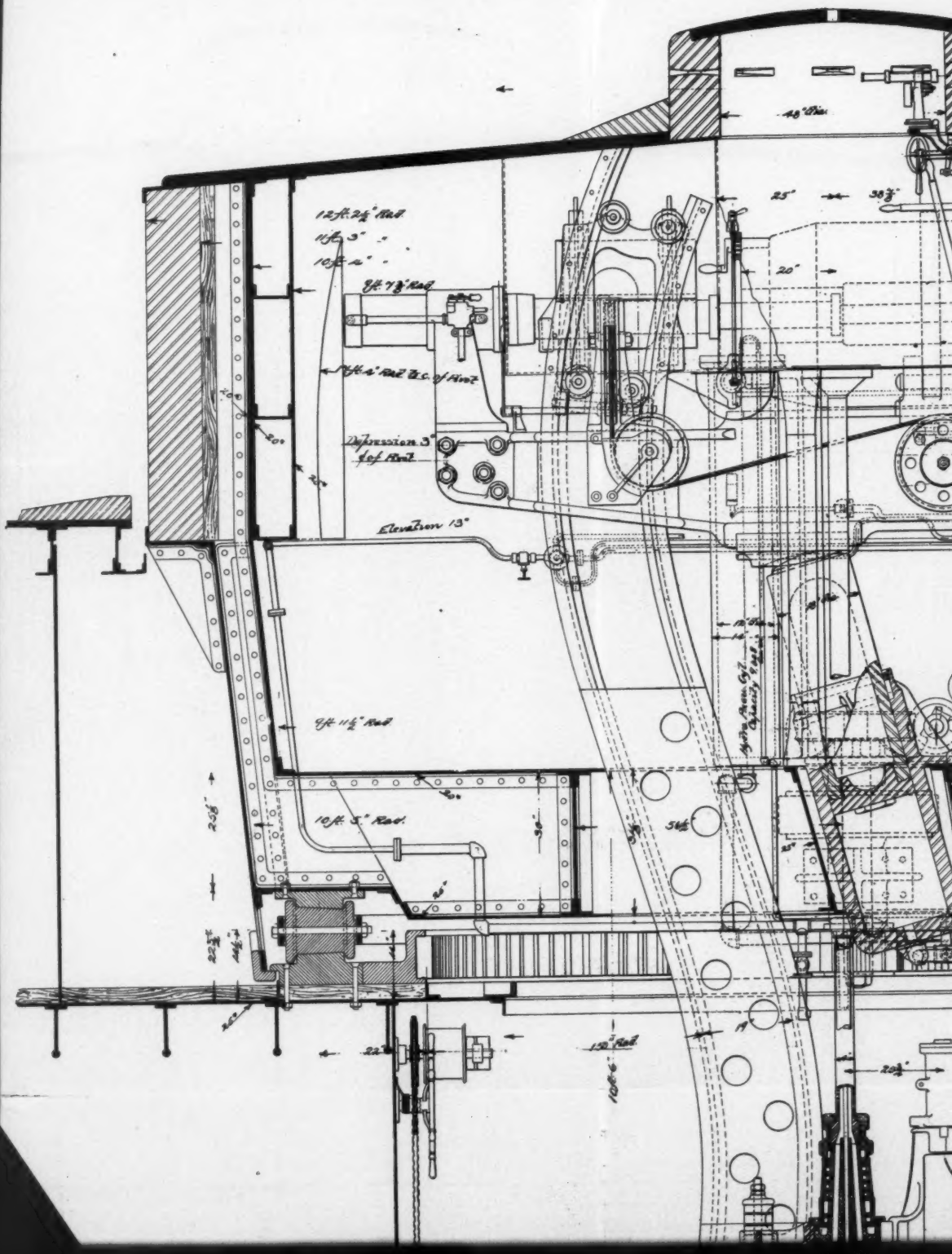
Recoil Cylinder.

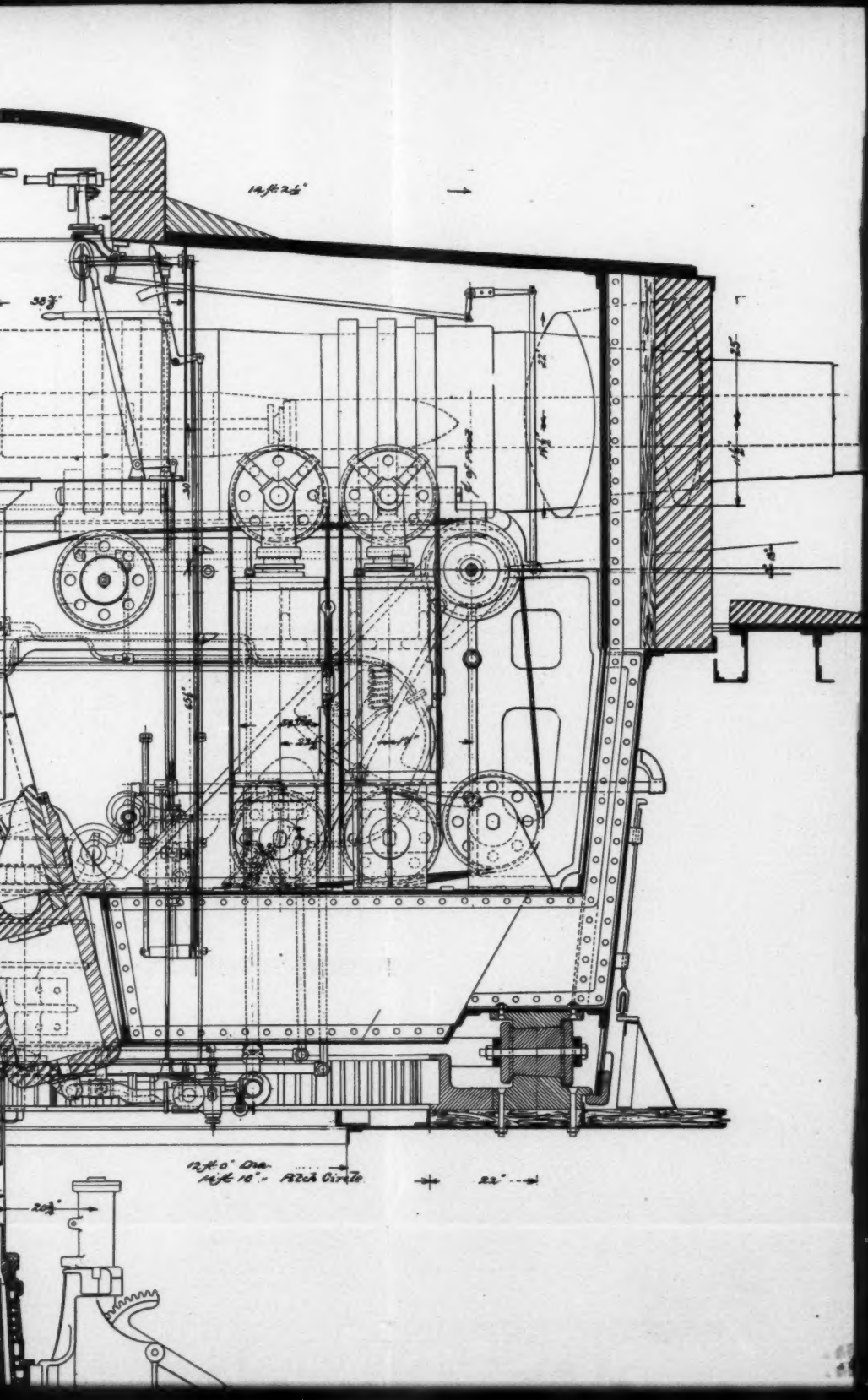
Patented

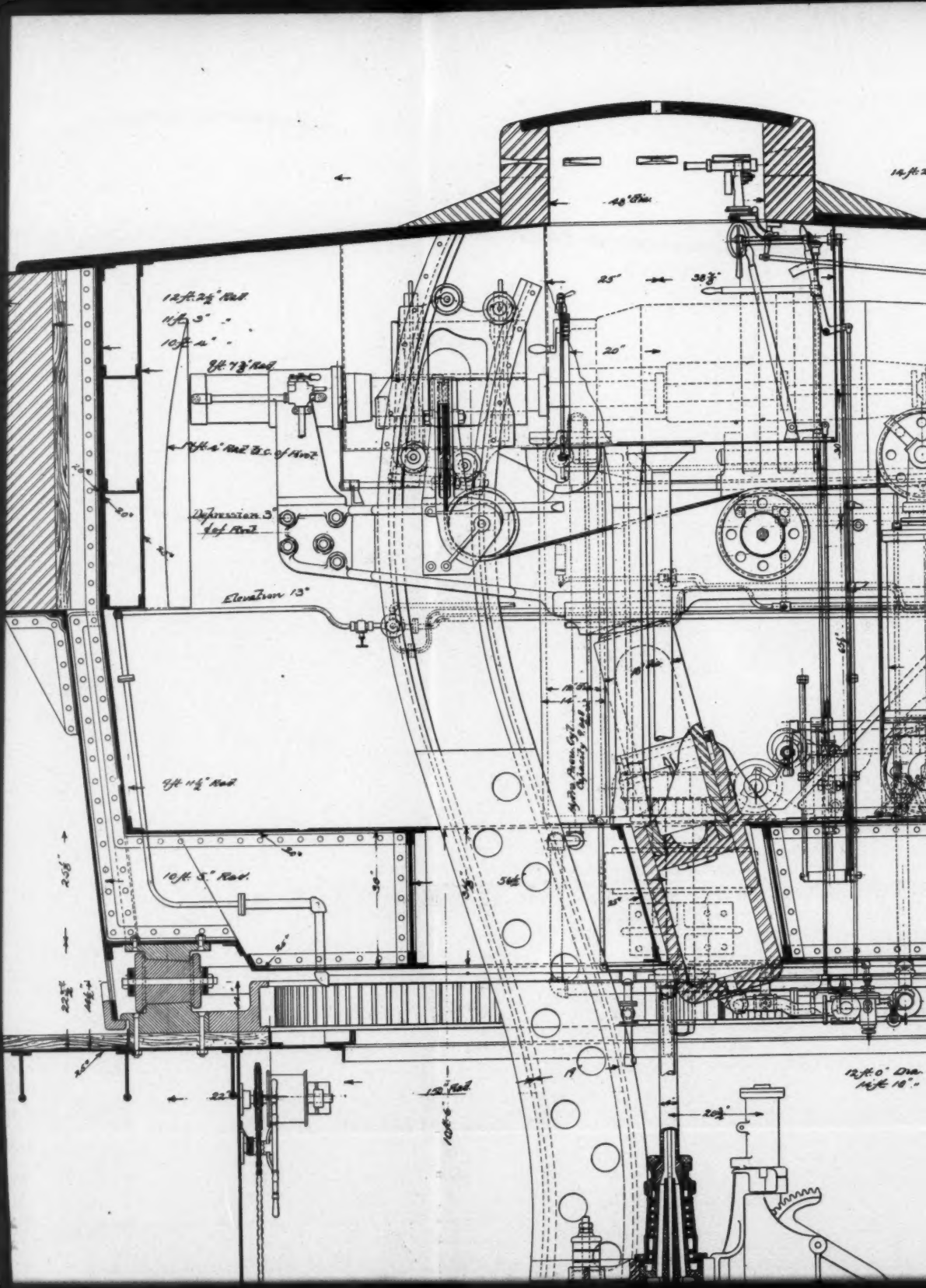
Pneumatic Gun Carriage and Power Co.

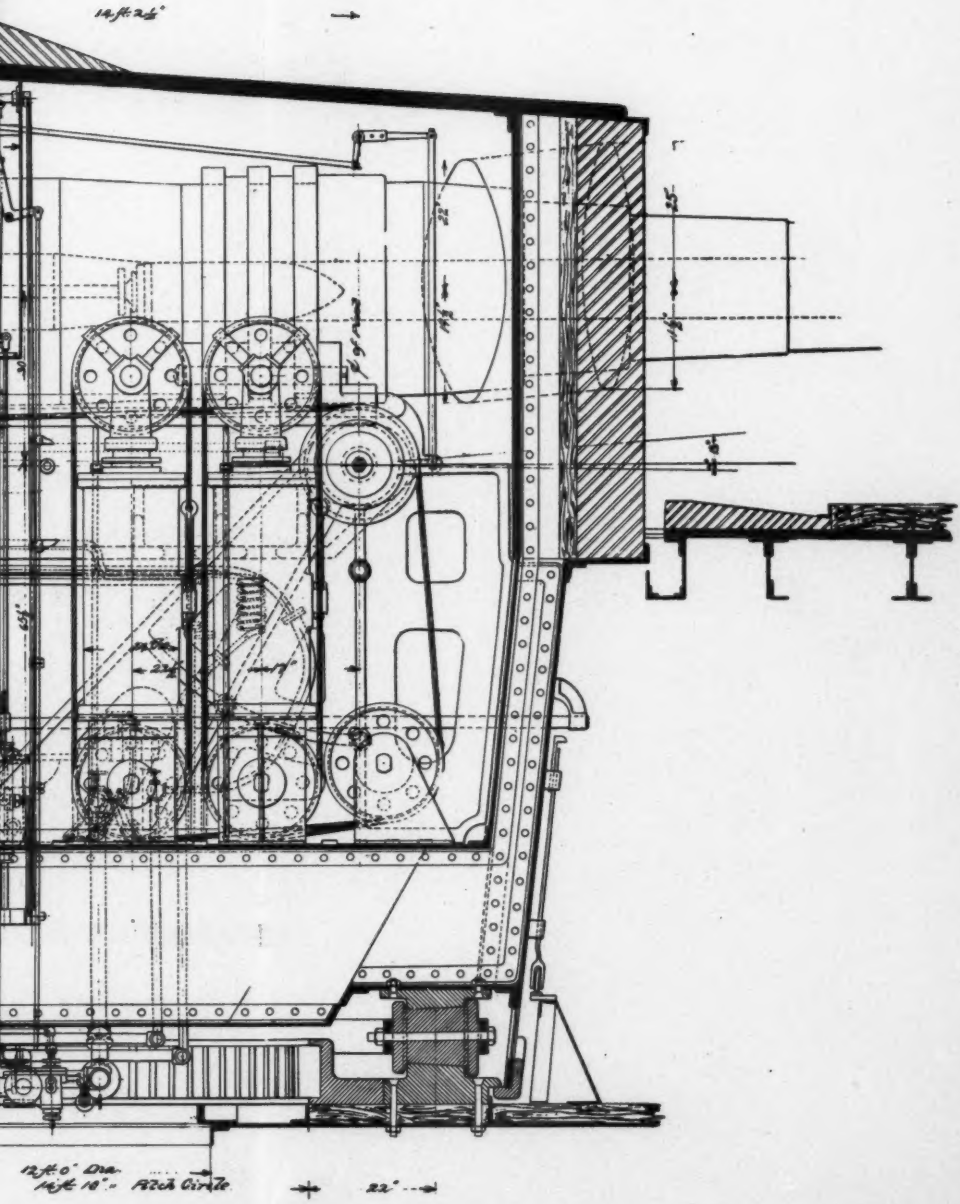


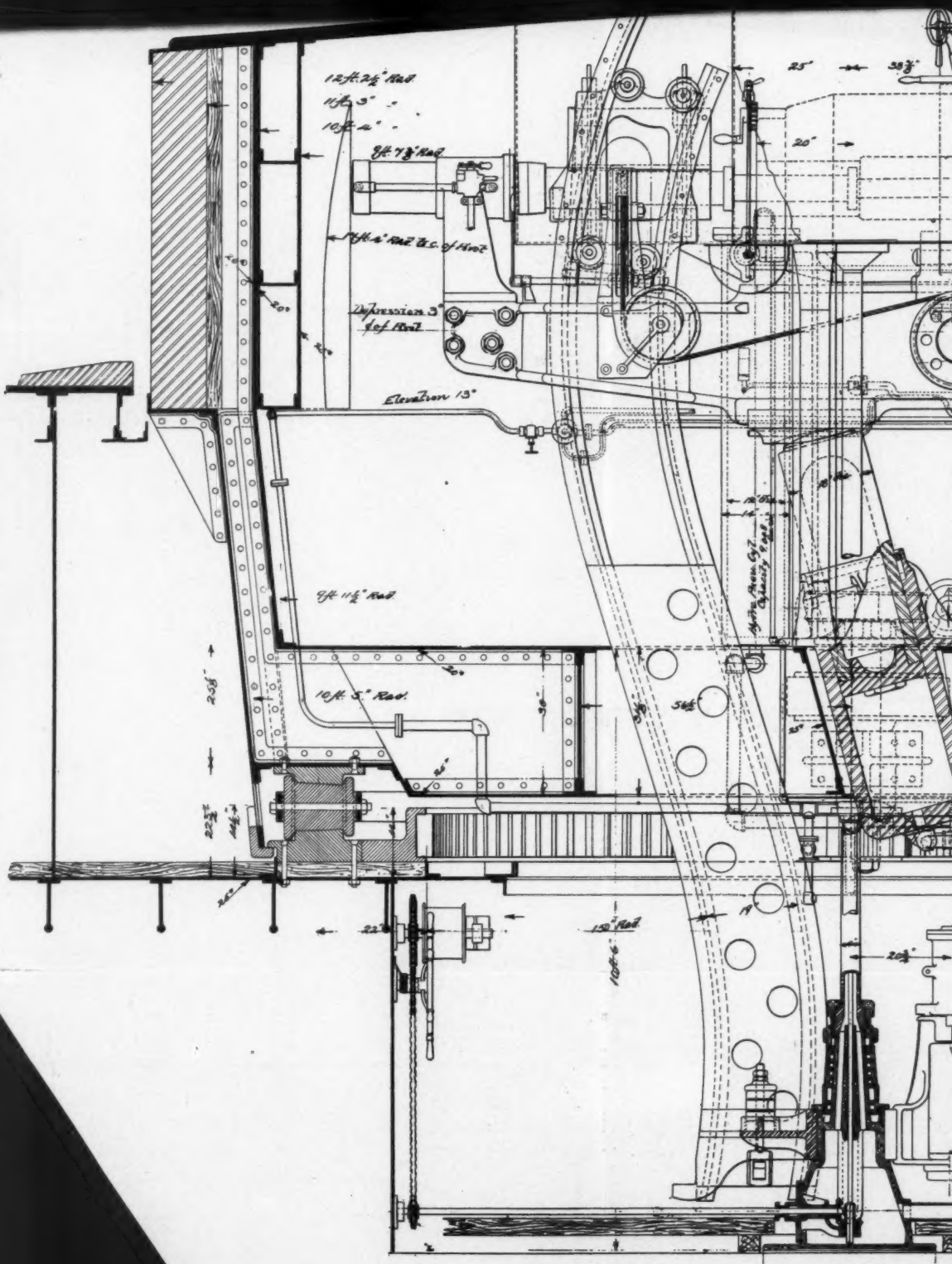

40°

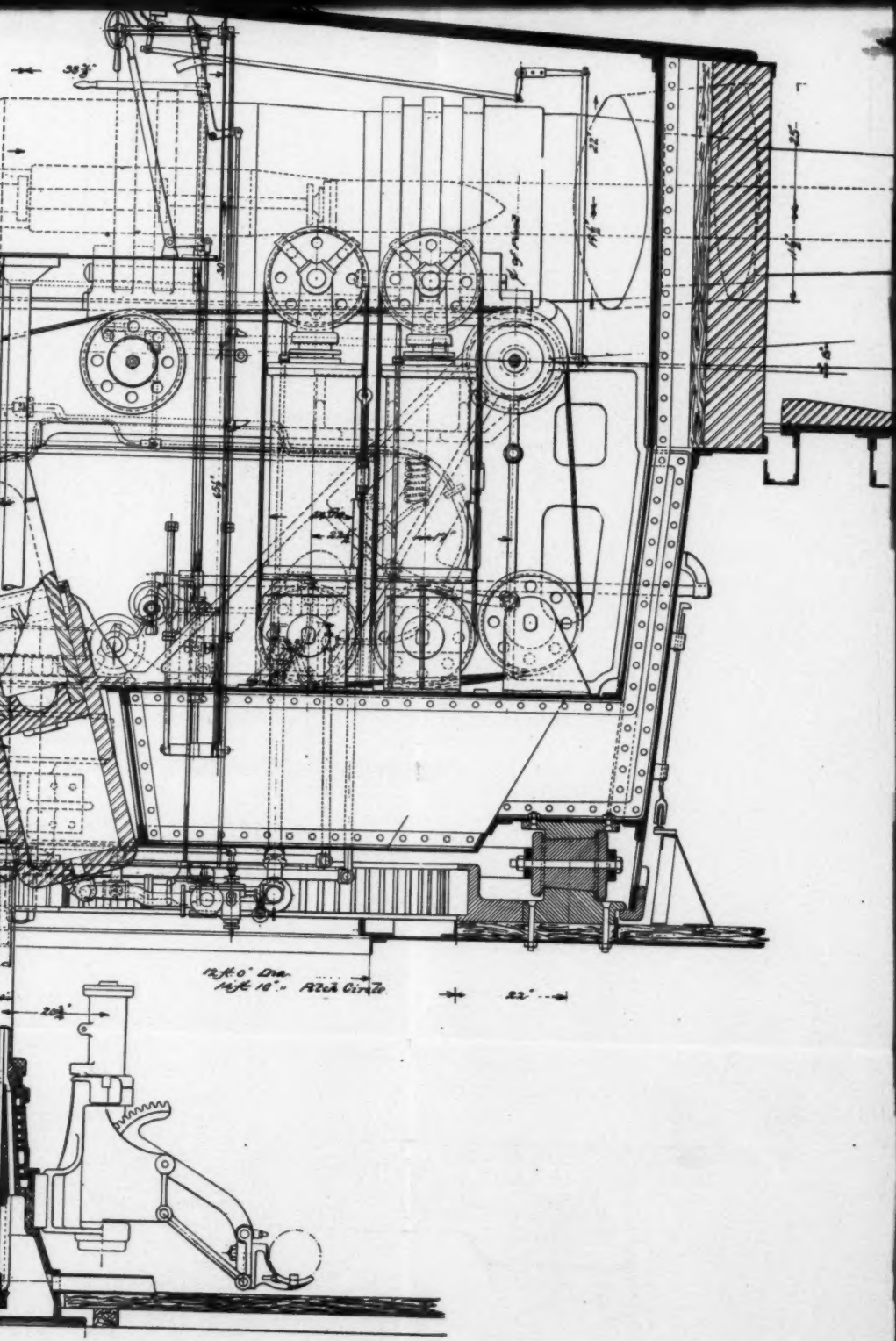

12ft. 2½" Rad
11ft. 3" "
10ft. 2" "
8ft. 7⅞" Rad
7ft. 4" Rad to C. of Rad
Depression 3°
Elevation 13°
7ft. 11½" Rad
10ft. 3" Rad.
12" Rad
10ft. 6"
49" Dia.
25"
30⅞"
20"
23⅝"
22½"
14½"
22"
20½"


14 ft. 2½"
38¾"
22"
14½"
25"
65"
12 ft. 0" Dia.
14 ft. 10" Arch Circle
22"
20½"


12 ft. 2½" Rad.
11 ft. 3" "
10 ft. 4" "
9 ft. 7½" Rad.
8 ft. 6" Rad. to C. of Pivot
Depression 3°
Elevation 13°
9 ft. 1½" Rad.
10 ft. 5" Rad.
12½" Rad.
12 ft. 0" Dia.
14 ft. 10"


14 ft. 2½"
12 ft. 0" Dia.
14 ft. 10" Arch Circle
22"


12ft. 2½" Rad.
11ft. 3" "
10ft. 2" "
9ft. 7½" Rad.
7ft. 4" Rad. to C. of Rod
Depression 3°
Elevation 13°
9ft. 11½" Rad.
10ft. 5" Rad.
150" Rad.
104.6
25"
39¾"
20"
23½"
22½"
22"
20½"


39½"
65"
22½"
12ft 0" Dia.
14ft 10" " Arch Circle
22"
20½"

THE SPEED PROBLEM.

By Mr. J.-A. Normand.

[Read before the Association Technique Maritime, Paris.]

Translated by Assistant Engineer R. D. Hasbrouck, U. S. N.

Having been invited by the Association to speak on the *Forban*, I find that there is no better way of replying to the invitation than in studying, in a general way, the question of the speed of torpedo boats. The *Forban* being only the last term of an increasing series, the law of the progress of speed will be easier to pick out if we consider the preceding terms.

The figures which I will give come from torpedo boats of my construction, because these are the only ones which are nearly exactly known to me.

It is principally in the small boats of 200 or 300 tons that the progress is manifest, and this because the phenomenon of resistance then presents a particular character which is impossible to discern in ships of great displacement.

This character consists in a rising due to the vertical component of the direct resistance, as I explained at our last meeting.

Let

D = displacement in tons.

V = speed in knots.

The direct resistance is proportional to $D^{\frac{2}{3}} V^2$, and this phenomenon of rising occurs when the ratio of the direct resistance to the displacement,

$$\frac{D^{\frac{2}{3}} V^2}{D} = \frac{V^2}{D^{\frac{1}{3}}}$$

attains a certain value.

In a torpedo boat of 125 tons, it is at about 20 knots that there is a minimum of utilization.

Above this the utilization increases, and we have in this case, calling V the speed of minimum utilization,

$$\frac{V_1^2}{D^{\frac{1}{3}}} = \frac{400}{5} = 80$$

The speed of minimum utilization is approximately given by the formula

$$V_1 = 9\sqrt[3]{D^2} \qquad (1)$$

For a vessel of 300 tons we have

$$V_1 = 9\sqrt[3]{6.695} = 23.3 \text{ knots},$$

and for a vessel of 8,000 tons,

$$V_1 = 9\sqrt[3]{20} = 40 \text{ knots}.$$

It is not probable that we shall live long enough to see the verification of the formula for a vessel of 8,000 tons.

The above figures are approximate, and should be corrected for the variations of the resistance with the absolute size; still the law appears certain.

The principal lesson which the *Forban* has taught us consists in having made evident the remarkable fact that the utilization increases gradually from 20 to 31 knots. We might fear that the slight increase in the utilization, observed at extreme speeds in all torpedo boats generally, might change sign for greater speeds, but the following table shows that this is not so.

Forban.

Speed, knots.	*Utilization.**
21.4	5.28
25.7	5.55
28.3	5.64
30.4	5.72

*These figures suppose a hull just out of dock, such as that of the *Forban* at her official trial. The progressive trials took place at different times, when the hull offered more resistance. However, the law of the increase of the utilization with the speed should be as exact as possible, considering the difficulty of measuring the power at very high speeds.

We see clearly, from what precedes, why it is that in the small vessels speed has made the greater progress.

This progress is considerable. Eighteen years ago the maximum speed was from 18 to 19 knots; in 1889, it barely attained 21 knots; in 1892, it suddenly jumped to 24 knots for the 80-ton torpedo boat and to 25.5 knots for the 120-ton boat. At last, after having slightly passed 27 knots on the *Chevalier*, it attains to-day more than 29 knots on the *Boxer*, 30 knots on the *Sokol* and 31 knots on the *Forban*.

We should deceive ourselves if we concluded that the speed had augmented, all other things being equal, 60 per cent. in this lapse of time. The above figures do not represent the curve of progress realized; they should be corrected for differences of absolute size and of load at trial. I shall try to deduce the real law. Very few questions are of such great interest as this one. Its scope is far from being limited to torpedo boats, and the lessons which come from it find application on the largest vessels.

Let

V = maximum speed.

F = maximum horse power.

D = displacement in tons.

a = fraction of displacement applied to the motive power complete, without spare parts or reserve supply of water, the importance of the latter being arbitrary.

π = weight per H.P. at maximum power.

M_D = corresponding co-efficient of utilization.

We have

$$V = M_D \sqrt[3]{\frac{F}{D^{\frac{2}{3}}}} \qquad (2)$$

and

$$\pi F = aD$$

$$F = \frac{aD}{\pi}$$

Substituting this value of F in equation (2), we have

$$V = M_D \sqrt[3]{\frac{a}{\pi} D^{\frac{1}{3}}} \qquad (3)$$

Formula (3) is as general as formula (2), and can be applied like (2) to all kinds of vessels. In a form, very little modified, this formula shows the causes of the progress realized.

It shows that the maximum speed depends

(1.) On the utilization.

(2.) On the fraction of total displacement employed in motive power.

(3.) On the weight per H.P. of the motive power.

(4.) On the displacement.

We will now examine successively the influence of these different elments.

Looking at the annexed table, which gives the results of different types of torpedo boats which I have built since 1886, we see that the utilization has not been bettered; it seems rather to have been diminished, but the reduction is probably only apparent. The length of the *Balny* was more than twelve times her width, while that of recent types is comprised between 9 and 9.5. The ratio of utilization to displacement has diminished, but the hull has become lighter, at least more solid, and at the same time the nautical qualities have been improved.

We would certainly obtain a notable amelioration of the utilization by suppressing, as has been done in most of the English torpedo boat destroyers, the vertical plan of the after part. This suppression, however, reduces in dangerous proportions the cruising stability while running at slow speeds in squadron and in a heavy sea.

However, it can be admitted that, between 50 and 200 tons, the influence of absolute size upon the utilization is approximately *nil.*

The advantage that follows an increase in displacement depends very much upon the arbitrary conditions of the problem. If we suppose, as I did in 1885 in my "Study of Torpedo Boats," that the equipment and all weights belonging to it, also the crew and water, increase only as the square of the linear dimensions, and that the weight of the armament, torpedoes and electric lighting plant, remains invariable, the improvement is considerable. But, in practice, this does not hold, because these weights vary more

nearly as the cube. When the absolute size is increased, it is done to obtain a greater radius of action and conditions of habitability compatible with longer cruises. The weight of armament and torpedoes also increases, attaining on the *Lancier* a figure twice that on a torpedo boat of 80 tons.

We should not be far from the truth in supposing that, if we vary only the absolute size, all other things being equal, the ratios of all weights to the displacement would generally vary very little. The values of M_D, a and π, in formula (3), being constant, the speed, therefore, varies as the ninth root of the displacement. By doubling the displacement of a torpedo boat of 20 knots, the speed becomes

$$20 \times \sqrt[3]{\sqrt[3]{2}} = 21.6 \text{ knots.}$$

Hence, it is not in an improvement of the utilization, nor in an increase of the absolute size that we must look for the reasons for the considerable progress realized in the last years. These reasons, however, appear conclusively when we examine the variations of the ratio a in the table. The cube root of this ratio is one of the coefficients of the speed, and the variations depend on those of the fraction a of the displacement that is given to the motive machinery, and above all on those of the weight π per horse power of the motive machinery. This weight, which was 72.6 pounds in 1889 on the *Avant Garde*, has been reduced to 35.2 pounds on the *Forban*, a reduction of 2 to 1 in six years. By examining formula (3), we see that this reduction alone increases the speed in the ratio of

$$\sqrt[3]{2} = 1.26 \text{ to } 1,$$

that is, for an initial speed of 20 knots,

$$20 \times .26 = 5.2 \text{ knots.}$$

This increase is, in reality, nearly 6 knots, if we take into account the amelioration of or improvement in the utilization at extreme speeds. It is this amelioration, together with an increase of about 20 per cent. of the fraction a of the displacement given to the

TORPEDO

Date of trials.....

Number of screws.....

Type of engine.....

Type of boilers.....

Boiler pressure, pounds per square inch.....

Weights on trials.

Hull, armed and equipped, with masts, awning and boats..... $= P$, tons.....

Motive machinery complete, engines and boilers with water, without spare pieces or reserve water..... $= P^{I}$, tons.....

Coal..... $= P^{II}$, tons.....

Artillery ammunition, torpedoes and tubes, air compressors and accessories..... $= P^{III}$, tons.....

Miscellaneous, completing the displacement on trials, equipment, provisions, drinking water, reserve water, electric lighting, and not otherwise accounted for..... $= P^{IV}$, tons.....

Displacement at trials equal to sum of above weights..... $= D$, tons.....

Weight of boilers alone, with water, grate bars, ash pans, masonry, fittings and tools..... $= P^{V}$, tons .....

Mean maximum speed on official trials..... $= V$, knots.....

Approxamative corresponding I.H.P..... $= F$.....

$\frac{P}{D}$ .....

$\frac{P^{I}}{D} = a$.....

$\frac{P^{II}}{D}$ .....

$\frac{P^{III}}{D}$ .....

$\frac{P^{IV}}{D}$ .....

Weight per I.H.P. of motive machinery complete..... $\frac{P^{I}}{F} = \pi$, tons..

Weight per I.H.P. of boilers alone..... $\frac{P^{V}}{F}$, tons.....

Factor of the cube of the speed..... $\frac{a}{\pi}$ .....

Radius of action at 10 knots corresponding to weight of coal on board at full speed official trials, deduced from coal consumption trials.....Nautical miles

Utilization, *i. e.*, value of M_D in formulæ $V = M_D \sqrt[3]{\frac{F}{D^{\frac{2}{3}}}} = M_D \sqrt[3]{\frac{a}{\pi} D^{\frac{2}{3}}}$..... $M_D =$ ...

BOATS.

Balny.	*Avant-Garde.*	*No. 128.*	*No. 140.**	*Lancier.*	*Chevalier.*	*No. 183.*	*Ariel.*	*Forban.*
1886.	1889.	1890.	1892.	1893.	1893.	1894.	1895.	1895.
1	2	1	1	2	2	1	2	2
Comp'd	Comp'd	Comp'd	Trip. Ex.	Trip. Ex	Triple	Triple	Triple	Triple
Loc'mt.	Loc'm't.	Loc'm't.	Normand	Du Tem.	Du Tem.	Normand	{Du Tem. Norm'd}	Norm'd
132.54	142.2	142.2	170.64	170.64	213.3	199.08	199.08	213.3
24.06	38.47	26.68	28.85	39.85	42.02	26.99	38.78	45.39
20.96	46.91	29.32	27.23	47.85	51.01	29.84	50.144	62.67
9.842	13.07	8.11	9.69	14.94	7.87	10.33	15.26	7.87
2.41	5.55	1.83	4.25	8.98	5.79	4.26	8.07	4.03
3.80	4.54	4.04	4.53	6.26	4.84	4.44	6.59	4.84
61.07	108.54	69.98	74.55	117.88	111.53	75.86	118.84	124.80
11.02	25.37	15.68	12.31	20.08	19.68	13.12	22.73	26.34
19.525	20.975	20.975	24.51	25.794	27.22	24.39	25.87	31.029
576.99	1,430.14	1,024.77	1,333.48	2,367.12	2,958.9	1,683.61	2,682.74	3,945.2
.394	.355	.382	.387	.338	.377	.356	.326	.363
.343	.432	.418	.365	.406	.457	.393	.422	.502
.161	.120	.116	.130	.127	.071	.136	.129	.063
.04	.051	.026	.057	.076	.052	.056	.068	.032
.062	.042	.058	.061	.053	.043	.059	.055	.04
.0364	.0328	.0286	.0204	.0202	.0172	.0177	.0187	.0159
.0191	.0178	.0153	.0093	.0085	.0067	.0078	.0089	.0066
9.42	13.18	14.60	17.85	20.03	26.4	22.15	22.50	31.6
1,690	1,800	1,800	1,800	2,000	1,140	2,190	2,030	1,240
5.86	5.27	5.34	5.7	5.57	5.4	5.35	5.37	5.74

* *No. 140* has given speed results greater, by more than half a knot, than the similar Normand torpedo boats delivered at the same time.

motive machinery, which explains the exceptional result of the *Forban*. This result, as we must recognize, has only been obtained by suppressing some weights, which, if not indispensible were at least useful, and transferring them to the motive machinery. Still it does not appear that the trials have been carried out with less displacement than those of the torpedo boat destroyers built in England. Therefore, as I explained last year, the weight per horse power of the motive machinery varies with the absolute size. Great as this weight is in machinery of small power, on account of the magnified importance of the auxiliary apparatus, the working tools, and the increased thicknesses necessary to allow for wear, it seems to reach a minimum in the sea-going torpedo boats and torpedo boat destroyers.

This minimum value can be reached on these boats, having between 4,000 and 6,000 I.H.P., only by giving a very high piston speed to the engines. The number of revolutions per minute on the *Sokol* was 420, while on the *Forban* it was only 365.

As the linear dimensions of the machinery and vessel increase, and as the service becomes longer and more important, so the specific weight of the engines increases as well as that of the boilers, in which the firing can not be kept up to a maximum for very long periods.

The large torpedo boats and the torpedo boat destroyers of 120 to 300 tons being the only ones considered in this paper, it can be admitted that the specific weight of the motive machinery is, therefore, independent of the absolute dimensions, and that it depends wholly, all other things being the same, on the economical working of the engines and boilers, and on the system of boilers.

The same observations are applicable to the hull, and within the same limits of displacement, the ratio of the weight to the cube of the linear dimensions is sensibly constant. For displacements greater than about 300 tons the ratio increases, provided that we do not wish to exceed a certain limit of strength in the plates.

It was on torpedo boats Nos. 126 to 129, and on the *Avant Garde* that I first applied feed water heaters, automatic drains, and cylinder valves discharging into the valve chests. Notwithstand-

ing that their engines were of the compound type, the economy of steam was considerable.

Later, on the first class torpedo boats No. 147, *et seq.*, and on the sea-going torpedo boats, type *Lancier*, the steam pressure was increased, the triple expansion system was adopted and water tube boilers were substituted for the locomotive type. The coal consumption diminished still more, and the total reduction in the weight of the boilers was equal to the sum of the reduction due to the change from locomotive to water tube boilers, the latter weighing about one-half as much as the former, (see table), and that resulting from a better utilization of the steam.

In 1886, the consumption of coal was about 2.64 pounds per I.H.P. at full power, and 2.2 pounds at low power, these figures being comparatively low. Owing to the various modifications mentioned above, the consumption has now dropped to 1.54 and 1.10 pounds respectively.

The reduction has been such that, notwithstanding that the radius of action imposed by trial requirements may have been increased, the absolute weight of combustible has diminished. This weight saved has served to increase the fraction of the displacement given to the motive machinery. Nothing could prove better the correctness of the principle, too long unknown, that the more economical the power, the lighter the machinery developing it.

The history of the progress of the torpedo boat is, therefore, that of the progress of the steam engine.

There only remains to examine what are the increases in the speed that we may foresee. It is indubitable that we are far from having reached the limit. The simple perfecting of means already known, the use of materials of greater resistance, of higher steam pressures, these alone would suffice to insure a notable increase.

We must forego, for the time being at least, the use of the light alloy of aluminum for the hull proper of sea-going vessels, on account of the galvanic action it sets up. This is regrettable, because when it is considered that the tension strains are very slight in hulls of small dimensions, and that a reduction of the scantlings is limited by the resistance to compression, and by

wear, it can be seen that an alloy of a density of 2.80, a tensile strength of 28,600 pounds per square inch, and an elongation of 16 per cent. would permit a reduction in the weight of the hull proper by about 40 per cent., or, say 20 per cent. of the hull armed and equipped.

But there is another cause of progress which has not yet been utilized and to which I wish to call your attention.

Actually, the volume of that part of the hull out of water is between one and one-half and two times that under water. Undoubtedly, a reserve of buoyancy is necessary, since water tight compartments are indispensible to safety, and it is important that the stability and navigability be assured when at least one of the compartments is filled. But is a volume of out-of-water body as great as is adopted at present indispensible? Yes, if the exterior service is to be performed upon the upper deck; no, if it is performed upon a raised platform, below which the sea can pass freely.

Between the submarine boat, where the reserve of buoyancy is *nil*, and the actual vessel, where it is greater than required by the volume of the water tight compartments and by the stability, and where it is important only in order to place the spar deck at a sufficient height, there exists a mean. The adoption of this mean will probably permit a great reduction in the ratio of the weight of the hull to the total displacement, which actually exceeds one-third, and consequently, an increase in the weight of the motive machinery. If it is possible to give the Whitehead torpedo a speed of 30 knots, it is because the complete immersion furnishes the displacement necessary for a powerful motor.

The *Forban's* dimensions are probably too much reduced to allow her to receive a raised deck advantageously. But let us see, without occupying ourselves with the changes in form or proportions necessary for resistance and stability, what the possible immersion, resulting from the addition of this superstructure, would be in this small vessel.

Let us suppose the immersion to be such that the displacement increases 50 per cent., or, from 126.8 to 190.2 tons. This hypothesis is within reason, for the volume of the hull out of water

reaches 220 cubic meters, or, say $1\frac{3}{4}$ times that of the submerged part under trial conditions.

It is also not inadmissible that, with a total volume of 340 cubic meters, even reduced in a great measure to fine the form, we can allow 185 cubic meters to the submerged part. The increase in displacement thus becomes 63.4 tons. From this we must deduct—

Raised bridge of gratings and accessories, ... tons,	4
Increase of coal necessary to preserve the same radius of action (the consumption per mile increases as the $\frac{2}{3}$ power of the displacement), ... tons,	2.5
Strengthening of the hull required by the increase in power, notwithstanding the great progress in balancing engines, ... tons,	2.5
Total, ... tons,	9.0

The increase of weight utilizable for the motive power thus becomes

$$63.4 - 9 = 54.4 \text{ tons},$$

and the total weight of the motive machinery, the weight of the *Forban's* machinery being 63.68 tons,

$$63.68 + 54.4 = 118.08 \text{ tons}.$$

The value of a will, therefore, be

$$\frac{118.08}{190.20} = .621$$

If the specific weight of the motive machinery and the utilization do not vary, the speed becomes—

$$V = M_D \sqrt[3]{\frac{F}{D^{\frac{2}{3}}}} = M_D \sqrt[3]{\frac{a}{\pi} D^{\frac{1}{3}}} = 5.74 \sqrt[3]{\frac{.621}{.0159}} \times \sqrt[3]{190.20}$$

$$\text{or, } V = 34.9 \text{ knots},$$

thus surpassing the actual speed by about 4 knots.

The utilization would probably not be diminished, since the speed of minimum utilization would be, approximately,

$$V_1 = 9 \sqrt[2]{\sqrt[3]{190.2}} = 21.6 \text{ knots},$$

that is, for the *Forban*, 13.3 instead of 11 knots lower than the maximum speed. The importance of the rising of the hull would then be considerable.

I hasten to add that in the case of the *Forban*, so great an increase in the speed cannot be realized without great changes in the motive machinery. The latter now occupies such a fraction of the interior volume that it cannot be increased. This would not be the case for larger vessels, and it seems probable that an increase of several knots could be obtained by this means, provided very high pressures were used in order to reduce the volume of the machinery.

Firing with liquid fuel would greatly facilitate the solution of the problem, because the length of the grates has reached a maximum compatible with the use of coal, and the power of the boilers is limited by the width of the grates that can be put into the hull.

It is needless to examine here the importance generally attributed to excessive speeds, a study of the question solely from a technical point of view being sufficient. But, if a raised deck is adopted, it will be more to improve qualities other than the speed, such, for instance, as the radius of action.

The free space between the raised deck and the hull would receive a reserve of coal in bags, excellent for protection, and which the use of water tube boilers would render dangerous in the fire room. In case of heavy weather, or when circumstances required the maximum speed, the vessel could be lightened rapidly. The transverse forms would be designed in a manner to assure the necessary stability with the extra weight in the superstructure. We could thus obtain a ratio of weights to displacement which would be inadmissible in an ordinary vessel having no water ballast, and could give to the armament and torpedo tubes a command as great as that on a vessel of three times the displacement.

Permit me to recall that sixteen years ago I proposed the raised platform for torpedo boat No. 60. This proposition was agreed to by the Council on Construction, provided that I increased the stability. But, as an increase in width would have prevented the realization of the required speed, the idea was abandoned.

Since then this arrangement has been applied with success in America upon certain "whalebacks."

Such are, if I am not mistaken, the causes of the recent progress, and some of the possible causes for future progress, in the speed of small vessels.

The principles are very simple, the application only being difficult.

In conclusion, I ought, perhaps, to excuse myself for having taken up so much of your time with vessels of apparently so little importance as torpedo boats; but importance is not always measured by absolute size. A celebrated foreign engineer has said that no other type has been the cause of greater progress in naval construction. Nothing is truer, and, apart from the bright light that this type has shed on the theory of the resistance of hulls and propellers, it has supplied the most valuable information on the steam engine.

The perfections applied to small vessels are applicable to the larger ones.

Whenever it is desirable, the largest engines will not require more than eleven pounds of water per horse power, and the weight of the boilers will be reduced to a small fraction of what it is to-day. When this progress has been accomplished, may the tiny vessel, which has shown the way to this realization, not be forgotten.

ON CRANK SHAFTS.

By John H. Macalpine, Esqr., Associate Member.

[Continued from page 726 of Volume VIII]

STRESSES DUE TO STEAM AND INERTIA.

We now come to the second part of the crank shaft problem.

The inertia of the piston and other parts having the same motion, attached to the crosshead *K*, Fig. V, produce a stress along the line of motion *O K*. This has a maximum upward intensity when the crank is at top center and, as the crank revolves, gradually falls to zero, which it reaches when the shaft has turned through rather less than a right angle. When the crank passes this point the inertia stress is downward and again reaches a maximum when the crank is at bottom center. The full line, Fig. X, shows exactly the law of this force when the connecting rod between centers is four times the crank radius and the engine is revolving uniformly. The dotted curve, which is a simple curve of sines, expresses the same law for an infinite connecting rod. These curves are laid out in the same manner as some of the previous diagrams. *AB* is the crank path expanded, the point *A* corresponding to the crank at top center, and the point *B* to the crank at bottom center. They repeat themselves in the reverse order for the second half of the revolution. In this and all subsequent inertia diagrams the curves extend a little beyond the top and bottom centers to include two points at which we will find maximum stresses in parts of the shaft occur. The ordinates are drawn for every 9° turned through by the crank. If the moving mass is one pound, the stroke one inch and the revolutions one thousand per minute, the end scale gives the inertia forces in pounds weight for an intensity of gravity equal to 32.2.

As the center of gravity, S, not shown in Fig. V, of the connecting rod, has both a vertical and horizontal movement, there are vertical and horizontal inertia forces induced which we may suppose applied at S. These vertical forces are of the same general character as those shown by the full line, Fig. X, only near A they are a little less intense per pound mass of connecting rod, and near B a little more intense. The horizontal forces through S are proportional to the displacement of S from the center line OK, that is to $\sin \theta$.

The angular motion of the connecting rod induces a couple which (neglecting friction) produces equal and opposite horizontal reactions from the main guide at K and crank pin L.

The inertia of the crank pin and webs produces a constant centrifugal force always directed along the crank radius. Its vertical component is consequently a curve of sines and is represented by the dotted curve, Fig. X. This radial force having no moment about the center of the shaft, the masses producing it have no effect on the turning moments on the journals. Thus the parts producing inertia turning moment are those more directly attached to the crosshead. The full line Fig. XI gives the turning moment produced by the inertia of a mass having the same motion as the crosshead and piston; the length of the connecting rod, as before, being four cranks. The end scale gives inch pounds turning moment per one pound moving mass per one inch stroke for one thousand revolutions per minute.

When the bearings are in line the maximum stresses occur at the same points of the shaft as when steam stresses alone are considered—that is, in the L.P. crank pin, after L.P. web, and aft L.P. journal where it joins the web. For these parts an examination of the various curves will make it evident, at least in the case of the aft journal and web, that when the engine runs so fast that steam and inertia combine to produce a greater stress than the maximum due to steam alone, these new maxima occur a little before the L.P. crank reaches top stroke. (The point of the revolution at which greatest stress occurs is not exactly the same for the various parts of the shaft.) For the steam and inertia stresses transmitted from K, Fig. V, are then both upward

in the L.P. engine and the inertia stresses from the L.P. connecting rod and crank are nearly so. Thus the cross bending forces arising from steam and inertia augment one another as also do the turning moments. The cross bending due to steam is falling (see Fig. VII), but that due to inertia is rising toward its principal maximum (see Figs. X and XII). The turning moment due to steam falls slightly for some time before top stroke is reached. The inertia turning moment in the aft journal (which is not very different from that due to the L.P. alone—shown by the full line in Fig. XI—since the I.P. and H.P. inertia turning moments oppose one another about this part of the revolution) is falling for about the last 30° of the upward motion of the crank. Thus, at the time the inertia turning moment passes its maximum near ordinate 4 *a*, Fig. XI, increasing crossbending stress from inertia is producing an increase in the total stress in the shaft, and therefore between this point and top stroke a maximum must occur, for just about top stroke there is a quick fall of stress due to the reversal of the steam pressure on the piston.

A similar set of conditions holds when the crank is nearing bottom center; but, as the inertia crossbending forces are seen to be feebler than those near top stroke, and the sum of the inertia turning moments from all three engines is negative, the maximum stress produced will be smaller.

When the L.P. engine is in the early part of the up or down stroke, the steam and inertia bending moments are of opposite sign and the combined stress is usually a good deal lower than near the end of the stroke.

The stress in the various parts of the shaft at all points of the revolution where a maximum could occur have been carefully calculated, and the results will be more fully noted when necessary.

As it is also found by trial that the L.P. crank pin has the greatest stress near top stroke we may confine our attention for the present to the last 30° or 40° of the upward revolution of the L.P. crank.

Let us now estimate the inertia turning moment on the aft main journal. The turning moment from the parts moving with the

TABLE XI.—DATA FROM ENGINES OF U. S. NAVY.

Line.	Column.	I.	II.	III.	IV.	V.	VI.	VII.	VIII.	IX.	X.	XI.	XII.	XIII.	XIV.	XV.	XVI.	XVII.
	Ship.	Weight of piston.	Weight of piston rod and crosshead.	Weight of connecting rod.	Weight of crank pin and half webs.	Weight of piston, piston rod, crosshead and half connecting rod.	Weight of piston, piston rod, crosshead and connecting rod.	Weight of piston, piston rod, crosshead, connecting rod, crank pin and half webs.	Ratio of I.P. weights in columns V and VII.	Ratio of I.P. weights in columns VI and VII.	Revolutions, per minute.		Piston speed, feet per minute.		$\mathfrak{M} = \frac{M' P^2}{400\, s^3 F_m}$		Mean surface velocity of crank pin, feet per minute.	
											Specified.	On trial.	At specified revolutions.	At trial revolutions.	At specified revolutions.	At trial revolutions.	At specified revolutions.	At trial revolutions.
(a)		H.P. 1,545 I.P. 3,186 L.P. 5,890	3,100	3,320	2,480	H.P. 6,305 I.P. 7,946 L.P. 10,650	L.P. 12,310	H.P. 10,445 I.P. 12,086 L.P. 14,790	...	...	110	116.25	770	814	...	...	446	472
	Baltimore.....																	
(b)	Ratios.....	.....	.....	.....	.....	H.P. .596 I.P. .746 L.P. 1.000	.....	H.P. .706 I.P. .817 L.P. 1.000	.720	.832	**1,117		...	...	4.24	4.74	...	...
(c)		H.P. 57 I.P. 128 L.P. 256	257	252	158	H.P. 440 I.P. 511 L.P. 639	L.P. 765	H.P. 724 I.P. 795 L.P. 923	...	...	240	222.4	800	741	...	...	377	349
	Bancroft.....																	
(d)	Ratios.....	.....	.....	.....	.....	H.P. .689 I.P. .800 L.P. 1.000	.....	H.P. .785 I.P. .862 L.P. 1.000	.692	.829	**.859		...	...	4.72	4.05	...	...
(e)		H.P. 1,450 I.P. 2,414 L.P. 5,849	3,443	6,489	2,810	H.P. 8,138 I.P. 9,102 L.P. 12,537	L.P. 15,781	H.P. 14,192 I.P. 15,156 L.P. 18,591	...	...	129	131.5	903	920	...	...	574	585
	Columbia.....																	
(f)	Ratios.....	.....	.....	.....	.....	H.P. .649 I.P. .726 L.P. 1.000	.....	H.P. .763 I.P. .815 L.P. 1.000	.674	.849	**1 039		...	...	6.38	6.62	...	...
(g)		H.P. 417 I.P. 698 L.P. 1,548	647	915	1,032	H.P. 1,522 I.P. 1,803 L.P. 2,653	L.P. 3,110	H.P. 3,011 I.P. 3,292 L.P. 4,142	...	...	†	††150.9	†	††754	...	...	†	††356
	Concord and *Bennington**..																	
(h)	Ratios.....	.....	.....	.....	.....	H.P. .574 I.P. .680 L.P. 1.000	.....	H.P. .727 I.P. .795 L.P. 1.000	.641	.751	.....		...	...	†	††5.01	...	...
(k)		H.P. 78 I.P. 125 L.P. 342	280	456	206	H.P. 586 I.P. 633 L.P. 850	L.P. 1,078	H.P. 1,020 I.P. 1,067 L.P. 1,284	...	...	200	¶218.8	800	¶875	...	...	367	¶401
	Mackias and *Castine*.																	

	Machias and *Castine*.																		
(l)	Ratios.....	.....	.....	.....	.....	H.P. .689 I.P. .745 L.P. 1 000	.....	H.P. .794 I.P. .830 L.P. 1.000	662	.840	**1 197		...	...	4.25	¶5.08	...	...
(m)		H.P. 481 I.P. 935 L.P. 1,975	999	1,565	702	H.P. 2,262 I.P. 2,716 L.P. 3,756	L.P. 4,539	H.P. 3,747 I.P. 4,201 L.P. 5,241	...	...	185	176.5	802	765	...	...	484	462
	Marblehead.....																		
(n)	Ratios.....	.....	.....	.....	.....	H.P. .602 I.P. 723 L.P. 1.000	.....	H.P. .715 I.P. .802 L.P. 1.000	.717	.866	**.910		...	...	5.50	5.01	...	...
(o)		H.P. 517 I.P. 1,171 L.P. 2,241	1,221	1,659	1,055	H.P. 2,568 I.P. 3,222 L.P. 4,292	L.P. 5,121	H.P. 4,452 I.P. 5,106 L.P. 6,176	...	...	150	162.1	750	810	...	...	432	467
	Monterey.....																		
(p)	Ratios.....	.....	.....	.....	.....	H.P. .598 I.P. .751 L.P. 1.000	.....	H.P. .721 I.P. .827 L.P. 1.000	.695	.829	**1.168		...	...	4.66	5.43	...	...
(q)		H.P. 1,127 I.P. 2,141 L.P. 4,299	1,951	2,873	2,640	H.P. 4.514 I.P. 5,528 L.P. 7,686	L.P. 9.123	H.P. 8,591 I.P. 9,605 L.P. 11,763	...	...	†	126.9	†	846	...	..	†	432
	*Newark**.....																		
(r)	Ratios.....	.....	.....	.....	.....	H.P. .587 I.P. .719 L.P. 1.000	.....	H.P. .730 I.P. .817 L.P. 1 000	.653	.776	.....		...	...	†	5 39	...	...
(s)		H.P. 795 I.P. 1,426 L.P. 3,006	1,975	3,233	1,735	H.P. 4,386 I.P. 5,017 L.P. 6,597	L.P. 8,214	H.P. 7,738 I.P. 8,369 L.P. 9,949	...	...	129	134.8	903	944	...	...	456	476
	New York (forward engine).																		
(t)	Ratios.....	.....	.....	.....	.....	H.P. .664 I.P. .760 L.P. 1.000	.....	H.P. .778 I.P. .841 L.P. 1.000	.663	.826	**1.092		...	...	5.70	6.23	...	...
(u)		H.P. 1,442 I.P. 2,909 L.P. 5,366	2,994	4,612	3,740	H.P. 6,742 I.P. 8,209 L.P. 10,666	L.P. 12,972	H.P. 12,788 I.P. 14,255 L.P. 16,712	...	...	†	119.5	†	797	...	...	†	469
	*Philadelphia**.....																		
(v)	Ratios.....	.....	.....	.....	.....	H.P. .632 I.P. .770 L.P. 1.000	.....	H.P. .765 I.P. .853 L.P. 1.000	.638	.776	.....		...	...	†	5 43	...	...

All weights are in pounds. The weights, except those in Column IV, are taken from the official weight sheets.

*The engines in these ships have built crank shafts.

**These ratios are the squares of the ratios of the revolutions in Column XI to those in Column X, and thus give the proportion in which the inertia stresses were increased or diminished at the trial revolutions from those at the specified revolutions.

† Revolutions not specified.

†† This figure is for the *Bennington*.

¶ This figure is for the *Castine*.

piston has already been given in Fig. XI; and it can be shown that, to an ample degree of approximation, the turning moment produced by the movement of the connecting rod may be replaced by that due to half the mass of the connecting rod moving with the piston. The inertia of the cranks being radial produces no turning moment.

Thus the masses to be taken as moving with the piston in estimating turning moment are the piston, piston rod, crosshead, and half the connecting rod. (If there are pumps or other gear worked off the crosshead, the mass of each working part diminished in proportion to the square of the ratio of its stroke to the stroke of the engine, should be added to the mass of the parts moving with the piston.) These masses are given for the ten engines of the U. S. Navy, previously used for determining constants, in Table XI, column V. In the same column, but in the lower line for each ship, are given the ratios which the weights for the H.P. and I. P. bear to the L.P. The mean value of the ratio H.P. / L.P., is .63, and the mean value of I.P. / L.P., .74. In no case do the respective ratios vary far from these values.

For any given values of the above ratios, the combined turning moment curve can at once be laid down, and this is done in Fig. XI, the dotted line giving the combined curve for order of cranks H.P., I.P., L.P., and the dot and dash line for the reverse order.

The ratios chosen in laying down these curves are I.P. / L.P. = .8 and H.P. / L.P. = .64, which give very slightly larger values for the dotted curve during the part of the revolution we are considering than if the mean values had been taken.

As it is near ordinate $3a$ that the maximum stresses occur when the bearings are in line; and as, for that position of the cranks, the L.P. inertia turning moment is much larger than that for the I.P. or H.P., a considerable variation in the choice of the above ratios does not materially affect the total moment.

It is also seen from Fig. XI that the order of cranks H.P., I.P., L.P., gives the largest turning moment near top stroke. The cross bending due to steam is also larger about this part of the revolu-

tion for this than for the reverse order of cranks, while that due to inertia is about the same for both orders. The turning moment in the aft journal due to steam is, of course, the same for both orders, and this is practically true also for the crank pin. Thus the shaft will be most severely stressed for the order of cranks H.P., I.P., L.P.; and to this we may confine our attention for the present.

The order of cranks H.P., I.P., L.P. will hereafter be referred to as the 1st order, and H.P., L.P., I.P. as the 2d order.

The parts to be taken in calculating crossbending are the piston, piston rod, crosshead, connecting rod, crank pin and about half the webs, the mean radius of the latter being only about half that of the crank. (Any gear worked from the crosshead should be added in the same way as noted in connection with the turning moments.) The weights of these with their ratios are given in Column VII of Table XI. Column VIII shows the ratio of the L.P. parts producing turning moment to those producing crossbending. In calculating the sizes of shafts this has been taken as .72, the highest value in Column VIII, so that in almost all cases the turning moments will be slightly overestimated.

Knowing all the inertia forces affecting each crank, the reaction at bearing *A*, Fig. IV, may of course be calculated by equation (14) when all the bearings are in line. The inertia due to the angular motion of the connecting rod is left out of account as the horizontal reaction it induces on the crank pin has a very small influence on the reaction at bearing *A* or on the crank pin itself. Omitting this allows us simply to add part of the weight of the rod to the parts moving with *K*, Fig. V, and the rest to the crank pin and half the webs. The inertia force of the first part is to be measured from the full line curve, Fig. X, and of the second part from the dotted curve. It will be practically accurate to make the same division of the L.P. moving masses as has been done in estimating inertia turning moment. Thus, if the total mass for the L.P. engine which produces crossbending is unity, we have the division of moving masses shown by Table XII.

TABLE XII.—DIVISION OF MOVING MASSES TO BE USED WHEN ESTIMATING INERTIA MOMENTS.

Line.	Engine.	L.P.	I.P.	H.P.
(a)	Mass to be taken as moving with the piston in estimating inertia turning moment and cross bending.....	.72	.72 × .8 = .58	.72 × .64 = .46
(b)	Mass which does not produce inertia turning moment, and which is to be taken as moving with the crank pin in estimating cross bending.....	.28	.28	.28
(c)	Total mass producing cross bending.....	1.00	.86	.74

NOTE.—Counterbalances on the cranks would diminish to an easily calculated amount the quantities in line (b).

As the inertia per pound of the parts moving with K and the vertical component from inertia per pound of the parts revolving with the crank are not greatly different, a considerable change in the above division of L.P. moving masses would not produce much difference in the estimation of the crossbending forces. As will be seen from Fig. XII, near ordinate $3a$, the inertia forces from the L.P. engine produce almost the whole crossbending, and therefore the exact value of the masses of the two other engines is a matter of comparative indifference.

In Fig. XII, the curves of reaction on the aft bearing are given. As these reactions are downward on the shaft (that is, according to convention, positive) when the L.P. piston is at top center, the ordinates at A must be reckoned positive. For clearness, the positive ordinates for the 1st order of cranks are laid off below, and for the 2d order above the base line AB. The curves X-X give the reaction arising from the proportion of the total masses given in line (a), Table XII, and the curves Y-Y that for line (b). The reaction for the L.P. engine alone is given by the dotted curves, that for the L.P. and I.P. by the dot and dash curves, and that for all three engines by the full line curves. The full line curve Z-Z, is the sum of the two full line curves X-X and Y-Y, and consequently gives the total vertical reaction.

Having seen that both in Figs. XI and XII the results are but little influenced by the determination of the moving masses of

the H.P. and I.P. engines near the point which we will find to be that of maximum stress, the great simplification of being able to estimate all moments with practical accuracy from the L.P. moving masses alone is reached.

Fig. XIII gives the turning moments on the L.P. crank pin. The full lines are for all bearings in line, the curves for the 1st and 2d orders of cranks being as marked. Moments are positive for the 1st order of cranks when the ordinates lie below the base line AB, and for the 2d order they are positive when above. The method of deducing the crank pin turning moment from the turning moment on the aft journal and reaction of the aft bearing has been fully explained. The curves have been deduced from those of the three previous figures on the supposition that the only part of the inertia forces of the moving masses and reaction at bearing A which modify the crank pin turning moment, are those arising from the masses attached to the cross-head given in line (a), Table XII. Denoting ordinate 2 of corresponding curves in Fig. X, XI, XII and XIII by the letters marked we then have

$$CD = EF + FG\left(1 - \frac{HK}{.72\ LM}\right).$$

Thus the curves of Fig. XIII can readily be deduced by a simple graphical process. The parts left out of account are those revolving with the crank pins. Of these the L.P. crank and parts directly attached have, obviously, no effect. If we could assume that equation (14) would apply to the reaction produced by the inertia of the parts revolving with the H.P. and I.P. cranks—the directions of which forces revolve with the shaft and, at the positions of maximum stress, make considerable angles with the resultant reaction of bearing A—it can be readily shown that the turning moments on the L.P. crank pin would have to be altered by about .6 of an inch pound per one inch stroke at one thousand revolutions and per one pound *revolving with the crank.* The crank pin turning moments would be increased by this amount for the 1st order of cranks and diminished for the 2d order. The alteration is slight and the application of equation (14) very doubtful.

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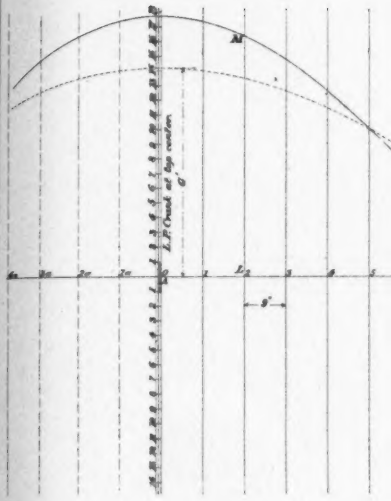

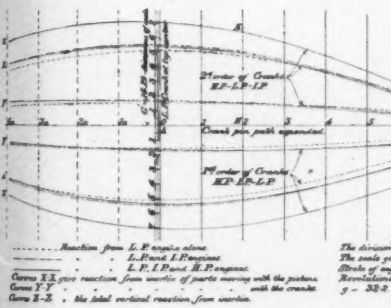

Reaction from L.P. engine alone.
L.P. and I.P. engines.
L.P. I.P. and H.P. engines.
Curve X-X, gives reaction from inertia of parts moving with the pistons.
Curve Y-Y . . . . . with the cranks.
Curve Z-Z . the total vertical reaction from inertia.

The divisions
The scale
Stroke of
Revolution
$g = 32.2$

FIG. X.

Curves of Inertia Forces.

The full line curve gives the inertia of mass moving with the crosshead. Connecting rod center to center = 4 cranks.

The dotted curve gives the vertical component of inertia of a mass revolving at the crank radius.

The scale gives the inertia force in pounds per one pound moving mass.

Stroke of engine = one inch.

Revolutions = 1000 per minute.

g = 32.2.

L.P. Crank at bottom center.

Crank pin path expanded. Direction of motion

g

FIG. XII.

Curves of Inertia Reaction on Aft Bearing A.

All bearings are in line. Connecting rod center to center 4 cranks.

Direction of motion

g

The division of masses moving with pistons and cranks is given on table XII.

The scale gives the reaction in pounds per one pound L.P. moving mass

Stroke of engine = one inch.

Revolutions = 1000 per minute.

g = 32.2

Forces are positive when downward on shaft.

Pth order of cranks. Ordinates below base line show positive reaction.

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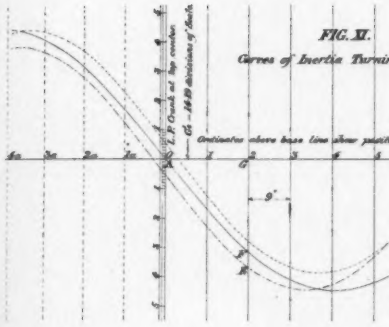


FIG. XI.

Curves of Inertia Turni

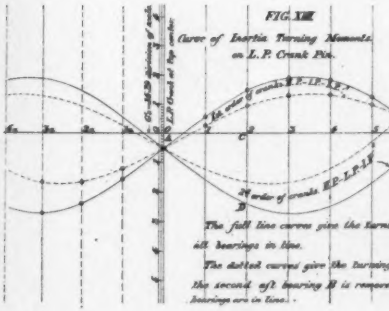
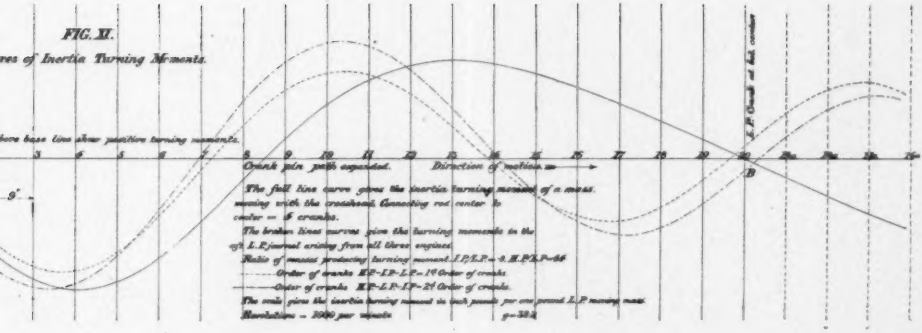

G. M.B. divisions of Scale.
L.P. Crank at top centre.
4 order of cranks E.P. I.P. L.P.
The full line curves give the turn
all bearings in line.
The dotted curves give the turning
the second aft bearing B is remov
bearings are in line.

FIG. XII.

Curve of Inertia Turning Moments
on L.P. Crank Pin.

FIG. XI.

ves of Inertia Turning Moments.



The full line curve gives the inertia turning moment of a mass moving with the crosshead, Connecting rod centre to centre = 6 cranks.

The broken line curves give the turning moments in the off L.P. journal arising from all three engines.

Ratio of masses producing turning moment I.P.S.P. = 4 H.P.S.P.=6.6

——— Order of cranks H.P.-I.P.-L.P.-1st Order of cranks.

— — — Order of cranks H.P.-I.P.-L.P.-2d Order of cranks.

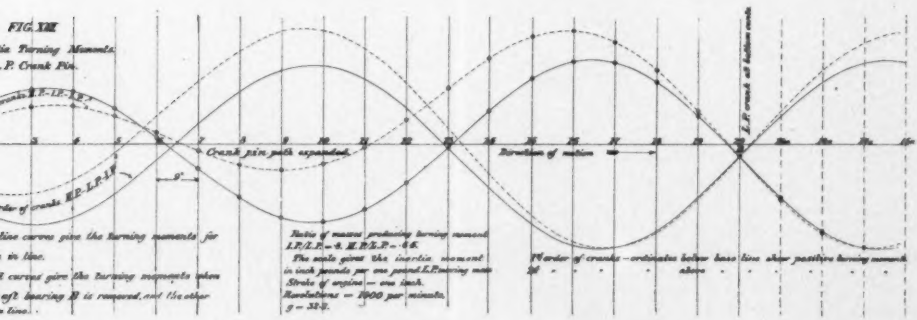
The scale gives the inertia turning moment in inch pounds per one pound L.P moving mass.

Revolutions = 2000 per minute. g=32.2

FIG. XII.

ia Turning Moments.

.P. Crank Pin.



Ratio of masses producing turning moment I.P.S.P. = 4, H.P.S.P. = 6.6.

The scale gives the inertia moment in inch pounds per one pound L.P moving mass

Stroke of engine — one inch.

Revolutions — 1900 per minute.

g = 32.2.

lines curves give the turning moments for th line.

curves give the turning moments when off. bearing B is removed, and the other e line.

H order of cranks — ordinates below base line show positive turning moments

2d " " " " above " " " "

Let us now suppose that the same engine is run successively at several speeds, but that the load is so adjusted that the steam distribution does not change; that is, that the same indicator diagrams are given at each speed. The change in loss from wire drawing and the action of the cylinder walls is of course neglected here. The curve of turning moments (Fig. III) and reaction f (Fig. VII) arising from steam pressure apply then in each case. But the inertia turning moments and reactions at bearing A given by Figs. XI, XII and XIII will have to be taken in proportion to the square of the speed of revolution.

If now we combine the sum of the bending moments from steam and inertia with the sum of the turning moments and thus calculate a curve of stress in, say, the crank pin for each speed of revolution, we shall find that the maximum stress does not occur at the same point for different speeds, but that as the speed increases it shifts toward the L.P. top stroke. This forms a slight difficulty in applying the same method for calculating the shafts when inertia is taken account of, as was used in deducing the curves of Fig. VIII; where the maxima, under the approximate suppositions made, always occurred at definite points of the revolution. The difficulty may be removed thus: It is not the point of maximum stress we wish to discover but the *value* of the maximum. The previous supposition with regard to the distribution of steam, moving masses, etc., being made, it was found by careful trial over the whole range of practice that when the stress due to steam and inertia is greater than that due to steam alone, the intensity of the maximum could be estimated with almost perfect accuracy for the crank pin by measuring the bending and turning moments for steam when the L.P. crank was 27° before top center, and measuring the inertia bending and turning moments when this angle was 22½°.

The same was found to apply in an equally satisfactory manner to the other parts of the shafts and to the measurement of the pressures on the bearings. The part due to steam was in all cases measured when the L.P. crank was 27° from top center but that due to inertia was taken at the points indicated in Tables XIV and XV.

It would take considerable space and be of little service to reproduce this part of the investigation as the errors avoided are in no case large, but the statements can be readily verified from the data given.

I will now give in Table XIII the estimation of the moments due to steam which are required in this part of the investigation. As in Table I, these are the values when all the bearings are in line. The remarks with regard to the various moments which precede the introduction of Table I and those referring to the various lines in that table apply here almost without alteration. Only the following need be noted:—

Line (d). The values in columns III and IV are really for a point a little less than 27° from top center; for, as noted in connection with Table I, cos φ is changing rapidly.

Line (f). The multipliers 1.06 in columns I and III are retained here though a 10 per cent. increase of the steam pressures in the L.P. cylinder would have a comparatively slight effect in raising the turning moment. But if the increase of pressure took place in one of the other cylinders, or was divided between one of these and the L.P., the case supposed in the table could readily be realized.

Line (k). As in Table I, the multipliers in Columns III and IV are scarcely reasonable. Their omission would correspond to a reduction of the breadth and thickness of the webs each by about 3 per cent. As the webs are so susceptible to rise of stress from want of alignment of the bearings, their retention would perhaps be justifiable.

We have now to estimate the moments due to inertia.

Let ω be the angular velocity of the crank in radians per second, M' the mass of the moving parts of the L.P. engine for stroke s, which are to be taken in estimating crossbending.

The crank radius in feet is $s/24$.

If the connecting rod were infinitely long the maximum upward inertia force would be

$$G' = \frac{M' \omega^2 s}{24 g}, \text{ in pounds weight.}$$

Let R be the revolutions per minute and P the piston speed in feet per minute. Then

$$P = \frac{2Rs}{12} \text{ or } R = \frac{6P}{s}.$$

$$\omega = \frac{2\pi R}{60} = \frac{\pi R}{30} = \frac{\pi \times 6P}{30s} = \frac{\pi P}{5s}.$$

Thus, we get

$$G' = \frac{M's}{24g} \times \frac{\pi^2 P^2}{25 s^2} = \frac{\pi^2}{24 \times 25 \times 32.2} \times \frac{M' P^2}{s} = .000511 M' P^2/s.$$

As $M' = Ms^3$ and P is supposed constant when we are considering similar engines,

$$G' = Gs^2.$$

The scales of Figs. X, XI, XII, and XIII are made for $M = 1$, $s = 1$, and $R = 1{,}000$. Thus $P = \frac{2{,}000}{12}$, and we have for the curves in these figures $G = .000511 \times \left(\frac{2{,}000}{12}\right)^2 = 14.19$ pounds weight.

Therefore for any value of M' and piston speed P, 14.19 divisions of the scale give the value of G', and we may at any point of the revolution measure the actual upward force at the center of the engine and the reaction on bearing A in terms of G' by dividing the scale reading at the point by 14.19, or by making a new scale which is changed in this proportion. In this way the values in line (d), columns II, IV and VI, Table XIV, are obtained.

Similarly, as 14.19 pounds exerted at an arm equal to the stroke of the engine would give a turning moment of 14.19 inch pounds per one inch stroke, dividing the scale reading for any point of Figs. XI and XIII by 14.19 and multiplying by $G's$ gives the turning moment in inch pounds, supposing the whole of M' to move with the crosshead. Hence the turning moments in line (d), columns I, III and V, Table XIV.

Line (d). It will be found that the bending moments in columns II, IV and VI are a few per cent. higher than the curves show for the first order of cranks. They are more nearly the values which these moments would have if the cylinders were arranged fore and aft in the order I.P., H.P., L.P. Occasionally the H.P. is in the center and it is perhaps well to cover this case. This arrangement of cylinders would lead to a smaller reaction due to steam than has been taken in deducing the bending moments in Table XIII, but the larger value must be retained if the bending moments are to be kept in excess at low as well as high speeds. The influence of this slight inconsistency on the subsequent determination of constants, Table XVI, and on the curves, Figs. XIV to XXI, is in all cases very small. The change in the crank pin turning moment from inertia is also so small that it has been entirely neglected.

In column III the turning moment is measured at the same point of the web as in Table I.

In column IV the factor for the crank and connecting rod being at an angle is .9 as before.

In column VI the moment has been measured as if the inertia of the webs and crank pin is exerted at the center line of the engine, *E*, Fig. IV. As it is somewhat distributed the moment in this column should be slightly lowered; but it can at once be shown that the effect is of no importance.

The supposition is, as in the corresponding part of Table I, that the pressure is not distributed over the brasses, so that here also the factor 1.05 in line (g), column VI, corresponds to an increase for errors of adjustment of almost 15 per cent.

Line (e). This multiplier is fully explained above.

Line (f). In calculating *G*, the value of *P* taken is supposed to be the intended piston speed. Frequently this will be exceeded, and consequently the inertia stresses will be raised in the square of the ratio of the actual to the intended revolutions. Columns X and XI, Table XI, give the specified and trial revolutions in the ten cases cited; and on the line below each of these, the square of their ratio.

It may be doubted if any allowance should be made for this

increase of stress; as, for each engine, the probability is that it will never be repeated after the full power trial and a slight increase of stress lasting only for a few hours will not cause sensible deterioration of the shaft. In Table XIV a ten per cent. increase has been made, and I think this is the very greatest allowance which could be justified by examination of the figures in Table XI.

This factor I have also considered ample to cover any rise of stress due to racing, since the engine will not be driven at near full power when there is danger of violent racing. As the admission of steam is always greatly restricted by throttling or linking up, in a very rough sea, a rapid increase of the revolutions always produces a correspondingly rapid fall of the steam pressures in the cylinders. Thus, though the inertia stresses may rise considerably, this rise is accompanied by a large drop in the stresses due to steam.

Of course the possibility of accident through the breakage of the after shafting or propeller—such as occurred in the *City of Paris*—is not contemplated here. The careful supervision of the engineer, or the adoption of some such mechanical device as has been proposed to prevent the revolutions rising above a safe maximum, can alone obviate this risk.

Lines (g), (h) and (k). The remarks appended to these same lines, Table I, apply here without change.

Line (l). The estimation from the drawings of the mass M' of the moving parts may readily, from several well known causes, fall below the actual: with care the error should be within 5 per cent. Hence the multiplier in this line.

These multipliers applied to the moments in line (d) give the moments in line (m).

Here also the shearing stress from torsion of the webs has in no case any influence on the maximum stress in the webs.

The total pressures on the crank pin and main journal, from steam and inertia, are given in Table XV; the values of the reactions at bearing A being for the 1st order of cranks.

The only multiplier which need be noticed is that in line (h) Column II. The inertia of the crank pin and webs does not con-

tribute to the pressure on the crank pin bearing. Consequently the ratio of the mass of the moving parts omitting the crank pin and webs, to the mass producing crossbending, has been given in Column IX, Table XI. The value of this multiplier adopted in Table XV is just under the highest value in Table XI.

The pressures in line (d), Columns II and IV, Table XV, are measured as if all the parts were moving with the crosshead, which gives a value very slightly in excess of the truth.

Using the values of the moments and pressures given in Tables XIII, XIV and XV, and the data in Tables III and XI, we find the stresses in the parts of the shafts and the surface pressures on the bearings which are given in Table XVI. As before, in estimating the pressure on the crank pin, the area of the bearing has been taken as the product of the diameter of pin and its length between the webs, instead of taking the product of the diameter and length of brass.

Instead of the ratio .72 in line (e), Table XIV, and the ratio .86 in line (h), Table XV, the actual values in Columns VIII and IX, Table XI, are used. Also the multipliers, 1.1 for excess of speed, and 1.05 for excess of weight in Tables XIV and XV, are cancelled, the actual speeds and masses being used.

The notes appended to the table will make the meaning of all the figures clear.

In calculating the stress in the round parts of the shaft we must combine the *sum* of the bending moments due to steam and inertia with the *sum* of the turning moments, to find the augmented moment. As the ratio U/P for steam alone and for inertia alone are different, the directions of the principal planes of stress, given by equation (4), for steam and inertia separately are not the same and consequently the two stresses are not additive quantities.

The ratios in Table XVI show to what extent the maximum stress due to steam and inertia exceeds that due to steam alone as given in Table IV. The objection may at once be raised that these ratios are all larger than unity and consequently it was not proper to determine the constants to be used, when steam stresses alone were considered, from these engines. The reply is simple.

If we had used the *Bancroft*, *Bennington* and *Castine* (that is, the three reference engines in which these ratios in Column I for trial revolutions are nearest unity) to determine the constants to be used, we would have deduced lower values than those in Table V, line (e). Therefore, the shafts calculated would have been heavier than we have found is justified by using the whole data. This anomaly is no doubt due to the fact that, probably, all or nearly all these shafts have had their diameters determined by a comparison with previous designs which took account only of the turning moments due to steam. Nearly all these engines have what would now be considered a moderate piston speed; and every day, engines of quite as high piston speed are having their shaft diameters determined by the ordinary formulas which, we have seen, only take partial account of the steam stresses. Again, it is not the object of this paper to tie down the designer in a hard and fast manner to particular constants, but rather to put the whole subject before him in such a way as will allow him to determine with the least trouble what dimensions are required for the constants he may think proper to adopt.

The adoption of higher constants in this second part is an illustration of the general principle that the more completely the facts are taken account of, the lower may be the nominal factor of safety allowed. After fuller investigation we can safely lower that part of the factor of safety which has very properly been called "the factor of ignorance"; and we are repaid for the additional care by avoiding on the one hand waste of material and additional cost, and on the other an unmeasured reduction of the actual strength and safety of the structure.

In Table XVII are given the average and maximum values of the stresses in Table XVI.

In line (g), Table XVII, are the constants which have been used in determining the shafts given by the full line curves in Figs. XIV to XXI, the values of the moments and pressures being those in Tables XIII, XIV and XV. The three assumptions given on pages 697 and 698, Volume VIII, are also made in these calculations. The crank pin pressure of 600 pounds per square inch is taken lower than would perhaps have been justi-

tribute to the pressure on the crank pin bearing. Consequently the ratio of the mass of the moving parts omitting the crank pin and webs, to the mass producing crossbending, has been given in Column IX, Table XI. The value of this multiplier adopted in Table XV is just under the highest value in Table XI.

The pressures in line (d), Columns II and IV, Table XV, are measured as if all the parts were moving with the crosshead, which gives a value very slightly in excess of the truth.

Using the values of the moments and pressures given in Tables XIII, XIV and XV, and the data in Tables III and XI, we find the stresses in the parts of the shafts and the surface pressures on the bearings which are given in Table XVI. As before, in estimating the pressure on the crank pin, the area of the bearing has been taken as the product of the diameter of pin and its length between the webs, instead of taking the product of the diameter and length of brass.

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The ratios in Table XVI show to what extent the maximum stress due to steam and inertia exceeds that due to steam alone as given in Table IV. The objection may at once be raised that these ratios are all larger than unity and consequently it was not proper to determine the constants to be used, when steam stresses alone were considered, from these engines. The reply is simple.

If we had used the *Bancroft, Bennington* and *Castine* (that is, the three reference engines in which these ratios in Column I for trial revolutions are nearest unity) to determine the constants to be used, we would have deduced lower values than those in Table V, line (e). Therefore, the shafts calculated would have been heavier than we have found is justified by using the whole data. This anomaly is no doubt due to the fact that, probably, all or nearly all these shafts have had their diameters determined by a comparison with previous designs which took account only of the turning moments due to steam. Nearly all these engines have what would now be considered a moderate piston speed; and every day, engines of quite as high piston speed are having their shaft diameters determined by the ordinary formulas which, we have seen, only take partial account of the steam stresses. Again, it is not the object of this paper to tie down the designer in a hard and fast manner to particular constants, but rather to put the whole subject before him in such a way as will allow him to determine with the least trouble what dimensions are required for the constants he may think proper to adopt.

The adoption of higher constants in this second part is an illustration of the general principle that the more completely the facts are taken account of, the lower may be the nominal factor of safety allowed. After fuller investigation we can safely lower that part of the factor of safety which has very properly been called "the factor of ignorance"; and we are repaid for the additional care by avoiding on the one hand waste of material and additional cost, and on the other an unmeasured reduction of the actual strength and safety of the structure.

In Table XVII are given the average and maximum values of the stresses in Table XVI.

In line (g), Table XVII, are the constants which have been used in determining the shafts given by the full line curves in Figs. XIV to XXI, the values of the moments and pressures being those in Tables XIII, XIV and XV. The three assumptions given on pages 697 and 698, Volume VIII, are also made in these calculations. The crank pin pressure of 600 pounds per square inch is taken lower than would perhaps have been justi-

fied by the data, and it should be satisfactory up to higher surface velocities than occur in these engines.

It is not proposed to enter fully here on the subject of lubrication, but the question is so vitally connected with the present investigation that a short notice of it seems to be necessary.

The minimizing of friction being one of the great factors in raising the efficiency of almost all machines, a very large amount of attention has necessarily been bestowed on the testing of lubricants chemically and mechanically. Till quite recently, however, the action of the lubricant in the bearing does not seem to have received any careful theoretical treatment.

In 1883–4 Mr. Beauchamp Tower made a set of experiments, already referred to, on the friction of a journal for the Institution of Mechanical Engineers, London. The journal was horizontal and was fitted with an upper brass only, which by means of a saddle, carried the applied load. By the slight angular displacement of this saddle the amount of friction was measured. Lubrication in the most important experiments was by means of an oil bath below the journal. The experiments were with nominal pressures ranging from 100 pounds per square inch up to the pressure at which the brass seized and from 100 to 450 revolutions per minute. These speeds of revolution correspond to surface speeds of 105 and 471 feet per minute respectively. There were also some experiments at very slow speeds.

For the detailed record of the measurements I must refer to the original paper. But Mr. Tower concluded that his experiments showed that journal friction, when the lubrication was perfect, follows the laws of fluid much more closely than those of solid friction; the friction being nearly independent of the pressure and increasing with the velocity, though at a slow rate. Increase of temperature had also a marked effect in diminishing friction.

He was led to measure the oil pressure at a number of points of the brass, and from these observations he constructed a series of curves showing the distribution of oil pressure between the journal and brass; thus clearly proving the presence of an oil film which established and maintained itself when the journal

was running under a load, so long as the lubricant was supplied below. His curves are reproduced in Fig. XXII.

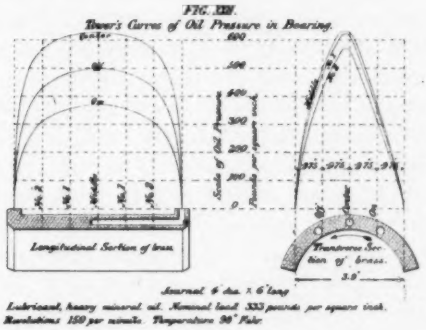


FIG. XXII.
Tower's Curves of Oil Pressure in Bearing.
Scale of Oil Pressure
Pounds per square inch
Longitudinal Section of brass.
Transverse Section of brass.
3.0"
Journal 6 dia. x 6" long
Lubricant, heavy mineral oil. Nominal load 333 pounds per square inch.
Revolutions 150 per minute. Temperature 96° Fahr.

In the longitudinal section the maximum pressure is shown at the center of the length of the brass. This he assumed, and measured the oil pressure on one side only of the center line. It is obviously correct if the brass is an equally good fit over the whole length and the load is central; and if, as in Mr. Tower's experiments, the bending moment at each section of the shaft is invariable. The pressure curves over a circumferential line are not quite symmetrical about the vertical line through the center of the shaft, but rise to a sharp maximum near this line and fall to atmospheric pressure at the edge of the brass.

Shortly after the publication of Mr. Tower's experiments the subject attracted the attention of Prof. Osborne Reynolds, who deduced the theory of lubrication from the known equations for the motion of a viscous fluid*. After developing the theory and explaining various physical considerations with regard to the generation of heat in the oil film, its escape by conduction and by the outflowing oil, the effect of change of load in changing the radius of the brass, etc., he proceeds to test his theory by applying it to some of Mr. Tower's results. The comparison of the calculated with the experimental values bears strong testimony both to the great accuracy of the experiments and to the suffi-

* See Phil. Trans. Royal Society, London, 1886.

ciency of the theory. As will be seen from an examination of Prof. Reynolds' Tables IV and V, his deductions and Mr. Tower's experiments follow one another closely over a large range of conditions.

Thus, in Mr. Tower's oil bath experiments, where the nominal pressure did not exceed 415 pounds per square inch—up to which pressure there was no liability to seize, at least when the journal had been running in one direction for some time—we have the most complete confirmation that the brass was perfectly oil borne and the varying thickness of the oil film is determined. The asymmetry of the curves in cross section, Fig. XXII, is shown to be necessary in order that the conditions of equilibrium of the brass may be fulfilled. The friction does not rise as the second power of the revolutions because the movement of the oil is non-eddying. The load of course tends to force down the brass on the journal so that they can never be concentric. There is thus a point in cross section of minimum thickness of the oil film the position of which is variable with the load and the velocity. As the speed of revolution increases under constant load the oil is drawn in more powerfully and it is shown that the brass and journal become more and more nearly central; that is, at the point of nearest approach the thickness of the film increases. This increase, together with a slight rise of temperature in the film and brass, prevent the friction rising at nearly the rate it would otherwise do. The viscosity of the oil at constant temperature, did no other conditions than the velocity change, would make the friction proportional to the velocity; but the fact that it varied in the oil bath experiments at only a little greater rate than the square root of the velocity at constant load, was due to the above noted changes in the thickness of the film and the temperature. Further, when this minimum thickness was less than about half of the difference of radii of the brass and journal, lubrication for Mr. Tower's brass becomes imperfect, as the oil film then falls to atmospheric pressure before the exit edge of the brass is reached. The investigation seemed to indicate clearly that this set the limit to the safe loading of the brass.

The above prepares us almost to anticipate the results of Mr.

Tower's experiments on other modes of lubrication. Introducing the oil by holes in the top of the journal, whether connected with a single groove running along the top of the brass, or with curved grooves extending partly over the circumference as well as lengthwise, was found to be impossible, and the brass invariably heated under a very light load. The grooves and oil holes obviously formed the best possible means of escape for the oil film as it was there reduced to atmospheric pressure. The pressure curves instead of being as in Fig. XXII would thus be divided, and the ordinates at the top of the brass instead of being nearly a maximum are reduced to zero. As the total upward pressure of the oil must equal the load, the maximum pressure in the film must be greater when the middle parts of the curve is thus reduced, and the pressure gradients become very much increased.

Oil was also introduced by cutting oil grooves at the sides of the brass parallel to the axis of the journal. These very considerably diminished the arc of contact of the brass and journal. This mode of lubrication was found to be entirely satisfactory if the load was not excessive. But the brass seized at 380 pounds per square inch while it had run well at a much higher pressure in oil bath experiments. This is readily accounted for, as the oil pressure gradients were necessarily greater when the effective arc of contact was diminished by cutting the grooves. The laws of friction in this case were similar to those with the oil bath, but the actual friction was a good deal greater—no doubt on account of the thinner oil film. In other experiments the oil was supplied below by means of an oily pad contained in a tin box full of rape oil. The oil flowed up by capillary attraction and gave so small a supply that the oiliness was just sensible to the touch. The resulting friction was usually much greater than with the oil bath. The seizing pressure was however practically the same in the two cases. This might be anticipated. For, however copious the supply of oil, as the load is increased toward that at which seizing will occur the brass is gradually drawn down and the oil film thinned. The amount of oil which can thus get through the journal is gradually diminished and, if the supply is kept equal

to this, the same results will be obtained as with a more copious supply. This result could not be expected to hold in journals where the pressure alternates every half revolution, as a copious supply of oil has then an opportunity of entering the brass when relieved of pressure. Thus the result applies only to the lubrication of crank shaft bearings in such cases as that supposed when calculating Table VIII.

The investigations spoken of above deserve and will repay the closest study and, so far as my knowledge extends, are by far the most philosophical writings on the subject. Perhaps the greatest advance they make is to firmly establish the presence of a sufficient mass of liquid to make the film subject to the ordinary laws of fluid motion, uninterfered with by those conditions which hold at the bounding surface of liquids and give rise to the phenomena of capillarity and surface viscosity. That the results of the experiments can also be deduced by use of the ordinary hydrodynamical equations puts this entirely beyond dispute. From this fact alone flows much which is important in the design of bearings of any description; and we see that, in many cases, the arrangements usually adopted for lubrication are far from being the best that could be made.

1st. The oil should be introduced and taken by the journal at a place where the brass has never to support the pressure of the journal—that is, between the top and bottom brass in crank shaft bearings.

This rule is not usually violated in those bearings where the pressure is always in one direction; simply because, as we have seen, the journal will not then take the oil or run cool except under very light loads.

It is almost invariably violated in those bearings where the force is reversed every half revolution, as in crank shaft bearings. Here the quantity of oil supplied is not checked, as all that passes down the oil hole flows in while the pressure is off the part where the hole enters. Thus, if the oil goes in at the top of the journal, the lower brass will be well lubricated; but when the pressure comes on to the top brass the oil pressure curves will be divided as we have already seen; the part of the

brass which should be most effective in bearing the load is rendered of no value; the pressure gradients are made steeper, so that, however quick the reversal of pressure, a thinner oil film will result. The film at the part of the bearing over which the journal runs in coming up to the oil hole will suffer especially, as the entering oil does not pass over it.

2d. When once introduced the lubricant should, as far as possible, be prevented from escaping from those parts of the brass which have to bear the greatest pressure, to the portions under less pressure.

This rule is usually violated if the first is; and further, the oil grooves which are always cut to distribute the entering oil over the bearing allow, when the pressure comes on the brass, outflow of oil from all the most important parts of the journal. The horse shoe thrust block is a case which may be cited in the present connection. The pressure on the surface is non-intermittent, and many thrust shoes have been made with no grooves, the oil being drawn in between the bearing surfaces from the trough below. This provision is not only sufficient, but I have no doubt is the best which can be made. When there is an oil box and oil grooves it is by no means unknown to have the oil flowing up into the oil box instead of down from it, but where this does not happen no doubt much of the oil escapes to that part of the groove which is first approached and is again drawn back between the surfaces at the other end of the groove. As the subject is of such great importance, I may be pardoned mentioning a case with which I had to deal some years ago. A vessel had been sent out with her thrust block supplied with the usual oil boxes connected with oil grooves extending over a considerable part of the bearing surface. The block gave much trouble from heating and there was considerable wear of the white metal. Guided by Mr. Tower's results I had the collars rebabbitted, omitting all oil holes and grooves, thus leaving the bearing surfaces quite flush all over, the oil being supplied by the collars dipping into the trough below, and the block afterward ran perfectly. The only alternative in explaining the effect is that the metal was not satisfactory in the first case. But

as only one quality of ingot was purchased there was not then, nor is there now, any suspicion in my own mind that this sensibly affected the result, even although it was a frequent practice to mix a little previously melted white metal with the fresh ingots.

Frequently the white metal is put into the journal in sections having recesses between; or grooves intended to distribute the oil not connected with oil holes, are cut. But any such grooves can, it seems to me, have only a harmful effect.

The pressure in a groove at any instant can never exceed the least pressure, at the same instant, in the whole surface through which it runs; and, if only partially filled with oil, may be at atmospheric pressure. Thus if run along the top of the brass, even for only a portion of the length, it will reduce the center curve in the longitudinal section, Fig. XXII, very considerably. If embracing a portion of the circumference its effect will be more marked, as the pressure falls much more quickly from the maximum in the cross-section curves than in the longitudinal.

But where the journal is subject to slight angular displacement from varying bending moments, as in the case of the crank shaft main journals, the effect of the presence of grooves must be considerably aggravated. Pressure curves for this case would usually be very different from those in Fig. XXII. The brasses will become very slightly rounded in the longitudinal section. When the pressure first comes on the lower brass of bearing *A*, Fig. IV, the point of maximum, when everything is in proper adjustment, will appear aft of the center line. As the pressure and consequent deflection increase, this point of highest pressure will shift to the forward side of the center line. The area of high pressure will necessarily be somewhat contracted, which implies a corresponding increase in the maximum pressure over that due to a journal with no varying flexure, and the pressure gradients will become larger in all directions from the point of maximum.

That this flexure, though slight, has an important influence on the action of the lubricant is, I think, strongly implied by the fact that crank pins have come to be run at a good deal higher

nominal pressure than main journals. No doubt the pressure of the main journal is more affected by errors of alignment than that of the crank pin, but this does not seem enough to account for present practice. The crank pin is placed at a portion of the shaft subject to much smaller angular displacements than the main journal, consequently the bearing will be more uniform and the endward escape of the oil more difficult. Again, it is usually found that the two outer main bearings wear most rapidly, and this is probably due to the fact that a center bearing gets its maximum pressure only when the force from the engines on each side is in the same direction and nearly equal; consequently the angular displacement of the intermediate journal is then small. The opposite is the case with the outer bearings. This angular displacement of the journals no doubt makes it more imperative that a heavy bodied oil should be used in a large machine while lighter oils can be used in similar small machines, though the nominal pressure is about the same in both cases; but even if there was no such displacement the large journal would require a somewhat heavier oil.

Professor Reynolds investigates the question of velocity of maximum bearing power and finds that with such perfect lubrication as was got with the oil bath, this velocity would have been from 1,500 to 2,000 revolutions per minute for Mr. Tower's journal (4 inches diameter.) This estimate took account of the outflow of heat produced by friction and the diminishing viscosity of the oil with elevation of temperature in the film, which have often formed a considerable obstacle in thinking on the subject. As this speed lay so far outside the highest revolutions in the experiments, no very definite figure could be arrived at. With very imperfect lubrication this speed may be indefinitely lower. But in crank shaft and other bearings, where the direction of the force is reversed every half revolution, the oil has a much better opportunity to enter than in Mr. Tower's journal, and probably the velocity of maximum bearing power is high even with the oil supplied in the usual way. The effectiveness of the oil supply is shown, especially in crank pin bearings, by the fact that they are often run at considerably

higher nominal pressures than Mr. Tower found to be possible with non-intermittent pressure. Taking account of these facts it was considered best to adopt constant nominal pressures in calculating the shafts in this investigation, and I would have little hesitation in retaining this value when designing an engine having $\mathfrak{M}$, the meaning of which is explained below, among the higher values used in calculating Figs. XIV to XXI. The mean surface velocity of the crank pin in the ten reference engines is given in columns XVI and XVII, Table XI.

Much that has been said above with regard to the introduction of the lubricants will be found very strongly urged in recent papers by Mr. Dewrance* after a very extended series of experiments; thus confirming the deductions which were made from the earlier investigations.

Other results of Mr. Dewrance's experiments are very well worth noting. "These experiments were originally instituted to demonstrate whether with similar lubrication and conditions a bearing surface composed of one alloy would allow of a greater load than a bearing surface of another alloy, and they have proved that it will not. The composition of the metal of the bearing surface has little or no influence on the load that the bearing will support. In no case was there the slightest evidence that this point (seizing) was reached sooner with one alloy than it was with another, so long as the metal itself would support the load. The experiments leave no doubt that so long as a bearing runs fairly cool, the surfaces of the shaft and bearing are separated by films of oil." The only advantage in lining the bearing is that the softer this metal the safer is the shaft from injury when seizing occurs. He advocates the compression test for alloys to be used, suggesting that before use it should be demonstrated that the point of first yield is considerably above the greatest load or shock to which it will be subjected in use.

He also notices the corrosive effect of the oil on the bearing surfaces; showing that oleic acid, the most common impurity of lubricating oils, corrodes lead and zinc rapidly, and copper less

*These papers are republished in the present issue of this JOURNAL.

rapidly, but that it does not attack tin and antimony appreciably. "Even if the oil is free from acid it becomes charged with oxygen from the atmosphere which oxidizes the surfaces, the oxide itself being immediately carried away by the oil." Finally, he recommends the use of an alloy composed principally of tin, and only containing enough of the most suitable metals to harden it to the utmost. It is possible to make an alloy of tin which will not yield under the compression test he prescribes until loaded to 8 tons per square inch.

Much useful information with regard to lubricants will be found in Prof. Thurston's "Treatise on Friction and Lost Work in Machinery and Mill Work."*

The horizontal scale in Figs. XIV to XXI is the value of F_m, and the vertical scale to the left gives the dimension of the part of the crank shaft to which the curve refers. Each curve is calculated for a particular value of the ratio G'/F'_m. As $G' = .000511\ M' P^2/s$ and $F'_m = F_m s^2$, each curve corresponds to a particular value of $\frac{M' P^2}{s^3 F_m}$. For brevity we may write

$$\mathfrak{M} = \frac{M' P^2}{400\, s^3 F_m} = \frac{M' P^2}{400\, s F'_m}.$$

The factor 400 in the denominator is introduced merely to make the numerical values of the ratio small.

The values of $\mathfrak{M}$ for the ten engines of the U. S. Navy are given in Table XI, columns XIV and XV, and it is at once seen that the curves cover much wider limits than occur in that table. Probably every engine running to-day, or likely to be built, has for $\mathfrak{M}$ a lower value than 10, though many are nearer it than the engines quoted here.

The curves for $\mathfrak{M} = 0$, 2, 5, 8 and 10 were calculated by trial and error and those for $\mathfrak{M} = 1$, 3, 4, 6, 7 and 9 interpolated.

There will, I think, be no difficulty found in using the curves, but for clearness and comparison with the results given in Fig. VIII an example is added.

Example III.—As in *Example I* take the engine cylinders =

* New York: John Wiley and Sons.

26 inches, 40 inches, 60 inches; stroke = 30 inches; boiler pressure = 160 pounds per square inch; as before, $F_m = 64.11$.

Suppose also that it is intended to run the engine at 190 revolutions per minute. Then $P = 950$. It will be remembered that a sufficient allowance has already been made in deducing the curves for the actual revolutions and piston speed exceeding the intended.

We have now to measure M' into which the weight of the crank pin, half the webs and the lower end of the connecting rod enter. These in turn depend on the dimensions of the crank shaft which we wish to determine. But from precedent, or by the use of Fig. VIII, the dimensions will be nearly enough known for determining this part of the total weight. If the shaft dimensions turn out considerably different from those assumed, a single revision of the work with the modified weights will certainly be sufficient. The weight of the piston and other parts can be directly estimated from the design.

Suppose $M' = 5{,}830$ pounds—nearly the value which would be deduced from the *Columbia*.

$$\mathfrak{M} = \frac{M' P^2}{400\, s^3 F_m} = \frac{5{,}830 \times 950^2}{400 \times 30^3 \times 64.11} = 7.60.$$

We have now to find the point on each diagram, the abscissa of which is given by $F_m = 64.11$ and which is above the curve $\mathfrak{M} = 7$ by .6 of the difference of the ordinates of the curves $\mathfrak{M} = 7$ and $\mathfrak{M} = 8$.

We get,

Part of shaft.	Figure.	Ordinate for $s = 1$, in inches—	Dimension of shaft = ordinate for $s = 1 \times$ stroke (= 30″), in inches.
Main journal, diameter	XIV	.364	10.92
length of bearing	XV	.551	16.53
Crank pin, diameter	XVI	.392	11.76
length between webs	XVII	.613	18 39
Crank web, breadth	XVIII	.431	12.93
thickness	XIX	.270	8.10
Span of main bearings, center to center	XX	1.741	52.23
Clear distance between bearings	XXI	1.190	35 70
Clear distance between web and main bearing brass	.....	.018+	.55

TABLE XIII.—MOMENTS DUE TO STEAM.

Line.	Column		I.	II.	III.	IV.	V.	VI.
(a)	Part of crank shaft		Aft main journal.		Aft crank web.		Aft crank pin.	
(b)	Moment		Turning.	Bending.	Turning.	Bending.	Turning.	Bending.
(c)	Point of revolution at which moment is measured		L.P. crank 27° before top center.					
(d)	Value of moment from Figs. III and VII		$.738\ F_m$	$.271\ F_m h$	$.701\ F_m$	$.244\ F_m k$	$.67\ F_m$	$.136\ F_m b$
(e)	Multipliers.	Ten per cent. loss from boiler pressure to absolute zero	9	·9	·9	9	·9	·9
(f)		Increase for unequal steam stresses	1.06	1.1	1.06	1.1	.....	1.1
(g)		Increase for errors of adjustment	.....	1.1	.....	1.1	.....	1.05
(h)		Shearing stress augmented in ratio 7 to 10	10/7	.....	.....	.....	10/7	.....
(k)		Increase for change of section	1.05	1.1	1.1	1.1	.....	.....
(l)	Value of moment taken in calculation		$1.056\ F_m$	$.325\ F_m h$	$736\ F_m$	$.292\ F_m k$	$.861\ F_m$	$.141\ F_m b$

The moments in Table XIII are for a stroke of one inch. The turning moments for stroke s will be got by changing F into $F_m's$ in columns I, III and V; and the bending moments by changing $F_m h$, $F_m k$ and $F_m b$ into $F'_m h'$, $F'_m k'$ and $F'_m b'$ in columns II, IV and VI, respectively. That is, all moments vary as s^2.

The bearings are supposed to be in line.

This table is for the 1st order of cranks—H.P., I.P., L.P.

TABLE XIV.—MOMENTS DUE TO INERTIA.

Line.		Column	I.	II.	III.	IV.	V	VI.
(a)		Part of crank shaft	Aft main journal.		Aft crank web.		Aft crank pin.	
(b)		Moment	Turning.	Bending	Turning.	Bending.	Turning.	Bending.
(c)		Point of revolution at which moment is measured	L.P. crank 27° before top center.		L.P. crank 22½° before top center.			
(d)		Value of moment from Figs. XI, XII and XIII	.312 G	442 $G\,k$	.285 G	.418 $G\,k$	.193 G	.234 $G\,b$
(e)	Multipliers.	Ratio of mass of parts producing turning, to those producing bending moment	72	.....	.72	.....	.72	.....
(f)		Increase for excess of speed	1.1	1.1	1.1	1.1	1.1	1.1
(g)		Increase for errors of adjustment	.....	1.1	.....	1.1	.....	1.05
(h)		Shearing stress augmented in ratio 7 to 10	10/7	.....	.....	.....	10/7	.....
(k)		Increase for change of section	1.05	1.1	1.1	1.1	.....	.....
(l)		Increase for moving masses being heavier than estimated	1.05	1.05	1.05	1.05	1.05	1.05
(m)		Value of moment taken in calculation	.389 G	.618 $G\,k$	.261 G	.584 $G\,k$	.229 G	.284 $G\,b$

The moments in Table XIV are for a stroke of one inch. The turning moment for stroke s will be got by changing G into $G's$ in columns I, III and V; and the bending moments by changing $G\,k$, $G\,k$ and $G\,b$ into $G'\,k'$, $G'\,k'$ and $G'\,b'$ in columns II, IV and VI, respectively. That is, all moments vary as s^2.

The bearings are supposed to be in line.

This table is for the 1st order of cranks—H.P., I.P., L.P.

TABLE XV.—TOTAL PRESSURES ON CRANK PIN AND MAIN JOURNAL FROM STEAM AND INERTIA.

Line.		Column	I.	II.	III.	IV.
(a)		Part of crank shaft	Aft crank pin.		Aft main bearing.	
(b)		Steam or inertia	Steam.	Inertia.	Steam.	Inertia.
(c)		Point of revolution at which pressure is measured	L.P. crank before top center. 27°	18°	27°	18°
(d)		Value of pressure from Figs. II, VII, X and XII	$.476\ F_m$	$1.156\ G$	$.271\ F_m$	$.486\ G$
(e)	Multipliers.	Ten per cent. loss from boiler pressure to absolute zero	.9	.....	.9	.....
(f)		Increase for unequal steam stresses	1.1	.....	1.1	.....
(g)		Increase for errors of adjustment	.....	.....	1.1	1.1
(h)		Ratio of mass of parts producing pressure on crank pin to those producing bending	.....	.86	.....	.....
(k)		Increase for excess of speed	.....	1.1	.....	1.1
(l)		Increase for moving masses being heavier than estimated	.....	1.05	.....	1.05
(m)		Value of total pressure taken in calculation	$.471\ F_m + 1.148\ G$		$.295\ F_m + .617\ G$	

The pressures in Table XV are for a stroke of one inch. The total pressures for stroke s will be got by changing F_m into F'_m and G into G'—that is, multiplying by s^2.

The bearings are supposed to be in line.

This table is for the 1st order of cranks.

TABLE XVI.—STRESSES DUE TO STEAM AND INERTIA COMBINED, DEDUCED FROM PREVIOUS DATA.

Line.	Column.	I.	II.	III.	IV.	V.
	Ship.	Augmented stress in aft L.P. main journal.	Stress in aft L.P. web.	Augmented stress in L.P. crank pin.	Pressure on main bearing.	Pressure on crank pin.
(a)	*Baltimore*	†14,984 15,540	15,349 16,025	14,066 14,747	174 186	648 697
(b)	Ratios	††1.21 1.25	1.23 1.28	1.20 1.26	1.45 1.55	1.32 1.42
(c)	*Bancroft*	11,811 11,368	10,781 10,276	11,652 11,108	384 349	606 549
(d)	Ratios	1.16 1.12	1.21 1.15	1.16 1.11	1.54 1.40	1.41 1.28
(e)	*Columbia*	13,583 13,748	17,303 17,583	12,823 13,048	444 456	676 695
(f)	Ratios	1.29 1.31	1.38 1.41	1.38 1.40	1.89 1.94	1.77 1.81
(g)	*Bennington**	 13,371	 8,796	 14,443	 262	 592
(h)	Ratios	..... 1.19	..... 1.26	..... 1.23	..... 1.61	..... 1.37
(k)	*Castine*	11,158 11,640	13,217 14,005	10,830 11,466	312 349	505 568
(l)	Ratios	1.13 1.17	1.16 1.23	1.12 1.18	1.44 1.62	1.33 1.49

(m)		13,985	14,841	15,647	439	626
		13,594	14,283	15,027	412	586
	Marblehead					
(n)	Ratios	1.25	1.34	1.30	1.71	1.61
		1.21	1.29	1.25	1.60	1.51
(o)		11,310	10,744	12,286	379	521
		11,813	11,377	13,075	419	578
	Monterey					
(p)	Ratios	1.17	1.23	1.20	1.53	1.40
		1.23	1.30	1.28	1.70	1.55
(q)		.....	.....	.....	.....	.....
		16,005	8,865	17,738	332	731
	*Newark**					
(r)	Ratios	.....	.....	.....	.....	.....
		1.25	1.33	1.29	1.69	1.47
(s)		12,480	15,059	13,150	360	584
		12,830	15,589	13,643	383	622
	New York (forward engine)					
(t)	Ratios	1.22	1.29	1.27	1.75	1.60
		1.26	1.34	1.32	1.86	1.70
(u)		.....	.....	.....	.....	.....
		15,705	8,564	15,847	326	702
	*Philadelphia**					
(v)	Ratios	.....	.....	.....	.....	.....
		1.30	1.36	1.32	1.70	1.48

The stresses in the metal and surface pressures are all in pounds per square inch. These quantities are constant for similar engines whatever the stroke.

* The engines in these ships have built crank shafts.

† The two upper lines for each engine give the stresses due to steam and inertia combined. The first of these two lines is the stress at specified revolutions, and the second that at the revolutions on trial. The intended revolutions are not specified for the *Bennington*, *Newark* and *Philadelphia*.

†† These ratios give the proportion of the stresses in the two upper lines of this table to the corresponding stress due to steam alone, which was given in Table IV.

TABLE XVII.—AVERAGE AND MAXIMUM VALUES OF STRESSES FROM TABLE XVI AND THOSE USED IN CALCULATIONS.

Line.	Column		I.	II.	III.	IV.	V.
(a)	Part of shaft		Aft L.P. main journal.	Aft L.P. web.	L.P. crank pin.	Aft L.P. main bearing.	L.P. crank pin.
(b)	Stress		Augmented stress in metal.			Surface pressure.	
(c)	At specified revolutions	Average stress	12,759	*13,899	12,922	356	595
(d)		Maximum stress	14,984	17,303	15,647	444	676
(e)	At revolutions on trial	Average stress	13,561	*14,163	14,014	347	632
(f)		Maximum stress	16,005	17,583	17,738	456	731
(g)	Values adopted in calculations		15,000	15,000	15,000	400	600

* The figures for built cranks in Table XVI are excluded in striking this average.
These values are all in pounds per square inch.

This example has been intentionally made as nearly as possible comparable with *Example I*, and the piston speed has been assumed comparatively high to show the absolute necessity of taking account of the inertia stresses if high speed engines are to be intelligently designed.

It is found that the shaft is $(10.92/10.35)^3 = 1.174$ times as strong as in *Example I* and the crank pin is $(11.76/10.50)^3 = 1.405$ times.

The webs, for resisting fracture from turning moments, are $\frac{12.93^2 \times 8.10}{11.55^2 \times 7.65} = 1.327$ times stronger.

For resisting fracture from bending moments they are

$$\frac{12.93 \times 8.10^2}{11.55 \times 7.65^2} = 1.255 \text{ times stronger.}$$

The areas of the main journals are increased in the proportion $\frac{10.92 \times 16.53}{10.35 \times 13.32} = 1.309$ times; and the area of the crank pin about $\frac{11.76 \times 18.39}{10.50 \times 12.39} = 1.662$ times.

Thus, in this high speed engine, if the crank pin had been made of the dimensions given in *Example I*, instead of having 440 pounds per square inch maximum pressure, it would have had nearly $600 \times 1.662 = 997$ pounds. The ratios in columns IV and V, Table XVI, show clearly that it is for the pressures on the bearings that the maximum for steam and inertia first exceeds that for steam alone.

The dotted curve in each of the Figs. XIV to XXI is deduced from the corresponding full line curve in Fig. VIII, by means of the transformation explained on page 725, Vol. VIII. It thus corresponds to the shaft in line. The stress in the metal of the shaft is 15,000 pounds per square inch and the surface pressures on the crank pin and main journal 574 and 287 pounds per square inch respectively. The surface pressures are lower than for the full line curves, but for steam and inertia the curves of reaction on these two bearings have a sharper maximum than for steam alone, and a somewhat higher pressure may be allowed

in the former case. Consequently the full and dotted curves may be considered fairly comparable. Probably no one would think it advisable to adopt a stress of 15,000 pounds per square inch when steam stresses alone are considered, unless for exceptionally strong steel. The dotted curve is introduced principally to show the points at which the maximum stress due to steam alone and that due to steam and inertia combined become equal, below which it would be entirely improper to use the full line curves.

I will now notice a point of great importance exhibited by the foregoing tables of moments and by the curves of shaft dimensions. In Table XVIII is given the ratio of bending to turning moments from Tables I, XIII and XIV.

TABLE XVIII.—RATIO OF BENDING TO TURNING MOMENTS FROM TABLES I, XIII AND XIV.

Line.	Column.		I.	II.	III.
	Part of shaft.		Aft L.P. main journal.	Aft L.P. crank web.	L.P. crank pin.
(a)	Steam alone. From Table I.		$\frac{.55\ k}{1.216} = .452\ k$	$\frac{.50\ k}{.85} = .588\ k$	$\frac{.25\ b}{1} = .25\ b$
(b)	Steam and inertia.	From Table XIII.	$\frac{.325\ k}{1.056} = .308\ k$	$\frac{.292\ k}{.736} = .397\ k$	$\frac{.141\ b}{.861} = .164\ b$
(c)	Steam and inertia.	From Table XIV.	$\frac{.618\ k}{.389} = 1.589\ k$	$\frac{.584\ k}{.261} = 2.238\ k$	$\frac{.284\ b}{.229} = 1.240\ b$

As we saw earlier, an examination of Fig. VIII showed that the presence of the bending moments caused the curves for the shaft dimensions to rise more quickly than in proportion to the cube root of F_m. The ratios of the bending to the turning moments for this case are given in line (a), Table XVIII. Line (b) shows the corresponding ratios for steam, when steam and inertia are both considered. Here the ratios are all smaller than in line (a). But the corresponding ratios for inertia given in line (c) are much larger than those in line (a). Taking a particular engine run successively at higher and higher revolutions, the actual ratios of the bending to the turning moments in the several parts of the

shafts, measured at the points noted in Tables XIII and XIV, will pass from those in line (b), which correspond to $P = 0$ and $\mathfrak{M} = 0$, toward the ratios in line (c); which, however, they will never quite reach. Thus an examination of the curves shows that, for the higher values of $\mathfrak{M}$, the dimensions of the shafts rise much faster than the cube root of F_m. If it increased in this ratio, the diameter of the crank pin for $F_m = 80$ would be double that for $F_m = 10$. For $\mathfrak{M} = 10$ and $F_m = 10$ the crank pin diameter is .193. For $\mathfrak{M} = 10$ and $F_m = 80$, instead of being $.193 \times 2 = .386$, it is .473 and the ratio of strength is $(.473/.386)^3 = 1.84$. $\mathfrak{M} = 10$ perhaps lies outside the limits of practice; but for $\mathfrak{M} = 7$, which is very well within these limits, we would have instead of 1.84, the ratio 1.62. The facts are even stronger than these figures appear to show. Suppose a number of engines are designed for, say, 900 feet piston speed each of which has the same boiler pressure, stroke and cylinder ratios but in which the corresponding cylinder diameters in the different engines are in the proportions $\sqrt{10}$, $\sqrt{20}$, $\sqrt{30}$, etc., so that the values of F_m will be successively 10, 20, 30, etc. Now consider the moving masses. The surface of the pistons will of course vary nearly as the diameters of cylinders squared, but the thickness will also increase so that the mass of the piston will rise more quickly than F_m. This rate will not usually be so fast as the cube of the cylinder diameters, for the parts lying near the circumference will generally increase more slowly than the piston body and there can be more economy of metal in a large than in a very small piston. The rate cannot be exactly stated, as it depends somewhat on the fancy of the designer. The piston rod and connecting rod shanks will increase approximately as F_m, but the parts about the crosshead and crank pin much faster if the surface pressures are kept constant. Thus on the whole, as we pass from $F_m = 10$ to higher values, we will also increase the ratio M / F_m and $\mathfrak{M}$; that is, we will increase the shaft dimensions more quickly than if we could pass along a particular curve as we supposed when ascertaining the ratios stated above.

Thus the bending moments, which we found for slow running engines should not be neglected, for high speed engines rise to a

position of great importance. Much valuable light might be thrown on the subject by a judicious analysis of data from fractured shafts; though the subject involves much labor and is greatly complicated by several considerations, such as the stiffness of the shafts and want of alignment of the bearings, accidental flaws in the material, &c., &c.

This leads to the remark that there is a limit to the weight per indicated horse power attainable by increase of revolutions which has, perhaps, not always been clearly enough appreciated. Rather than discuss the matter fully I will give the simplest illustration. Suppose that by means of a crank and relatively long connecting rod we give a reciprocating motion, which will be practically simple harmonic, to a uniform steel rod of one square inch section and 16 feet long, the travel being one foot. The weight of the rod is about 55 pounds. Suppose any tendency to buckle is prevented and we wish the load per square inch not to exceed 5,000 pounds. When the revolutions are 250 per minute the stress due to inertia alone will be 585 pounds per square inch, and it will be allowable to exert a force on the free end, as the steam presses on the piston, of $5{,}000 - 585 = 4{,}415$ pounds. By doubling the sectional area and consequently the weight of the rod we can exert another 4,415 pounds at the free end, and so on. When the revolutions reach 500 per minute the force it will be allowable to exert at the free end of the rod of one square inch section will be $5{,}000 - 4 \times 585 = 2{,}660$ pounds, and by doubling the section and weight of the rod we now only gain an additional 2,660 pounds. When the revolutions have reached 731 per minute the whole allowable stress of 5,000 pounds per square inch is exerted by inertia and no increase of section would allow us to exert any force at the free end. The maximum work per pound of rod would be reached at much lower revolutions. If the force exerted at the free end consisted of a constant pull and push always exerted in the direction of motion of the rod, it can easily be shown that the maximum work exerted per pound weight of rod would be at 422 revolutions per minute.

If the illustration had included the crank actuating the rod the result would have been even more striking, but the discus-

sion would have been too complex. All through high speed machinery there are similar limits existing. Thus for any one character of design (say an engine of any particular type with given stroke, cylinder diameters and steam pressure) and definite factors of safety, it is impossible to build a machine to run faster than a perfectly definite speed; and the economical speed, measured by work done per pound weight of machine, will always lie far below this limit. Probably in many cases of high speed engines lightness has been obtained by an unintentional lowering of the factor of safety.

As is no doubt well known, in two similar engines, run at the same piston speed and having the same boiler pressure, the weight per indicated horse power is directly proportional to the linear dimensions. (This follows at once from the proposition at the beginning of the paper. For the indicated horse power will be proportional to the cylinder areas—that is, to the squares of the linear dimensions—while the masses, moving and stationary, are proportional to the third power.) This will also be true for the economical speed, as the stresses in the material of the two engines being the same at the same piston speed the economical speed with a given factor of safety must always correspond to one piston speed whatever the linear dimensions; and the limiting speed spoken of above, beyond which an engine of definite type and definite factors of safety cannot be built to run, will correspond to a definite piston speed also.

The inertia stresses are obviously cumulative as we pass from the piston to the crank shaft, so that the speed of the engine for which the maximum stress from steam and inertia is greater than that from steam alone, is higher for the parts above the crank shaft than for the crank shaft itself. Probably no engine ever runs so fast as to have the maximum stress in the piston higher than that due to steam alone. The consideration of inertia stresses will then require most addition to the lower parts of the moving masses and will consequently tend to decrease slightly the ratio of the parts producing turning to those producing crossbending. But the crossbending becomes so important in high speed engines that, as can easily be verified, a small

reduction of this ratio will produce a much smaller proportionate reduction in the stress even in the aft main journal, where cross-bending is smaller than in the parts immediately forward of it.

It is also very important to notice that this cumulative effect of inertia forms a powerful protection to the crank shaft against shocks from water or other obstruction in the cylinder. The brunt of the blow is taken by the upper portions of the moving parts, whose inertia has to be overcome; the slack of the main journal, crank pin and crosshead bearings being then always so disposed that it must be reversed before the crank shaft takes any of the shock. If the blow is not central the elasticity of the piston rod and other parts also increases the protection of the shaft. It is a matter of common experience that in almost all cases of breakage from such causes the upper moving and stationary portions suffer most.

(To be continued.)

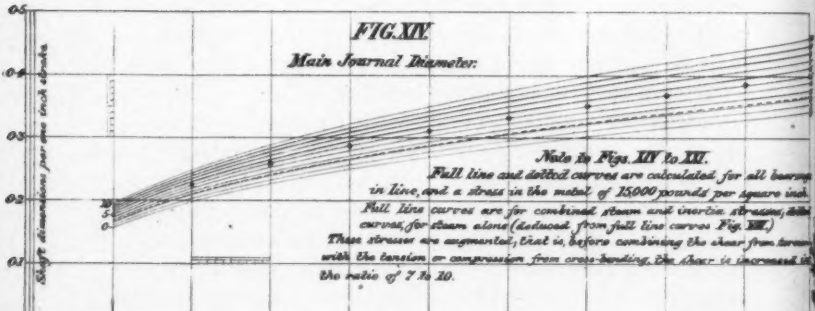


FIG. XIV.
Main Journal Diameter.
Shaft dimensions per one inch stroke.
Note to Figs. XIV to XIV.
Full line and dotted curves are calculated for all bearings in line, and a stress in the metal of 15,000 pounds per square inch.
Full line curves are for combined steam and inertia stresses; dotted curves, for steam alone (deduced from full line curves Fig. XIII.)
These stresses are augmented; that is, before combining the shear from torsion with the tension or compression from cross-bending, the shear is increased in the ratio of 7 to 10.
Scale of F_m.

TABLE OF MULTIPLIERS.

F_m	50	60	70	80	90	100
10	1.015	1.030	1.045	1.058	1.070	1.082
9	1.005	1.022	1.037	1.050	1.061	1.072
8	...	1.013	1.028	1.041	1.052	1.063
7	...	1.005	1.018	1.031	1.042	1.052
6	...	...	1.009	1.021	1.032	1.041
5	...	...	1.001	1.012	1.022	1.031
4	...	...	...	1.004	1.013	1.022
3	...	...	...	...	1.004	1.013

Second aft bearing, B, removed and the other bearings in line—augmented stress in metal = 15,500 pounds per square inch.

Multiplying the ordinates of the full line curves by the multipliers in this table corresponding to the same values of F_m and $\mathfrak{M}$ will give the required diameter of main journal.

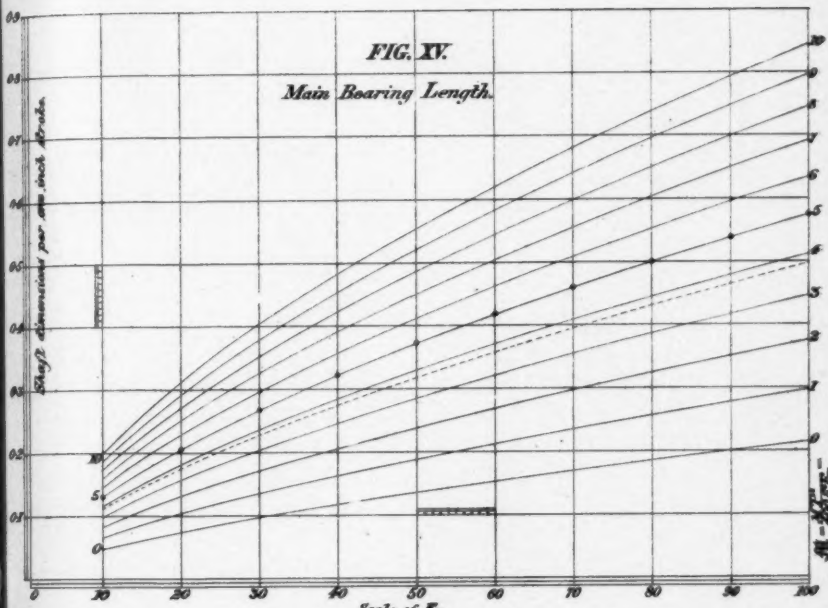


FIG. XV.

Main Bearing Length.

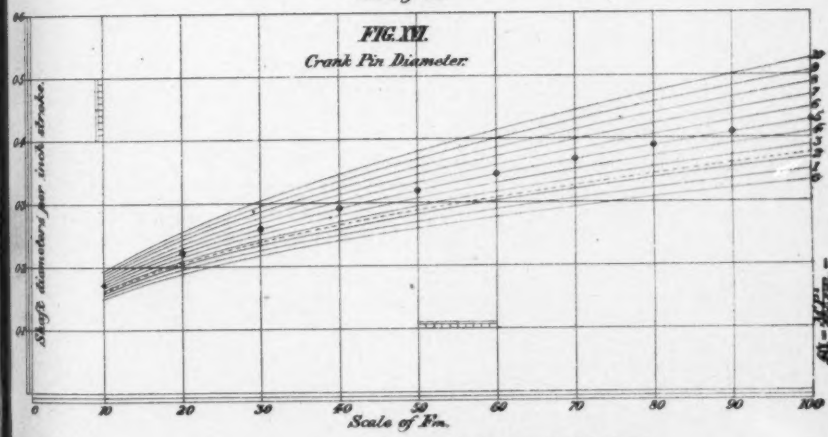


FIG. XVI.

Crank Pin Diameter.

TABLE OF MULTIPLIERS.

Second aft bearing, *B*, removed
nd the other bearings in line
-augmented stress in metal =
,500 pounds per square inch.
Multiplying the ordinates of the
ll line curves by the multipliers
this table corresponding to the
ame values of F_m and (B) will give
e required diameter of crank
in.

F_m	40	50	60	70	Bo	90	100
10	1.003	1.009	1.013	1.016	1.019	1.022	1.024
9	...	1.006	1.010	1.014	1.017	1.020	1.022
8	...	1.003	1.008	1.012	1.015	1.018	1.020
7	...	1.001	1.005	1.009	1.012	1.015	1.017
6	...	...	1.002	1.007	1.010	1.012	1.015
5	...	...	...	1.004	1.007	1.009	1.012
4	...	...	...	...	1.004	1.006	1.009
3	...	...	...	...	...	1.003	1.005

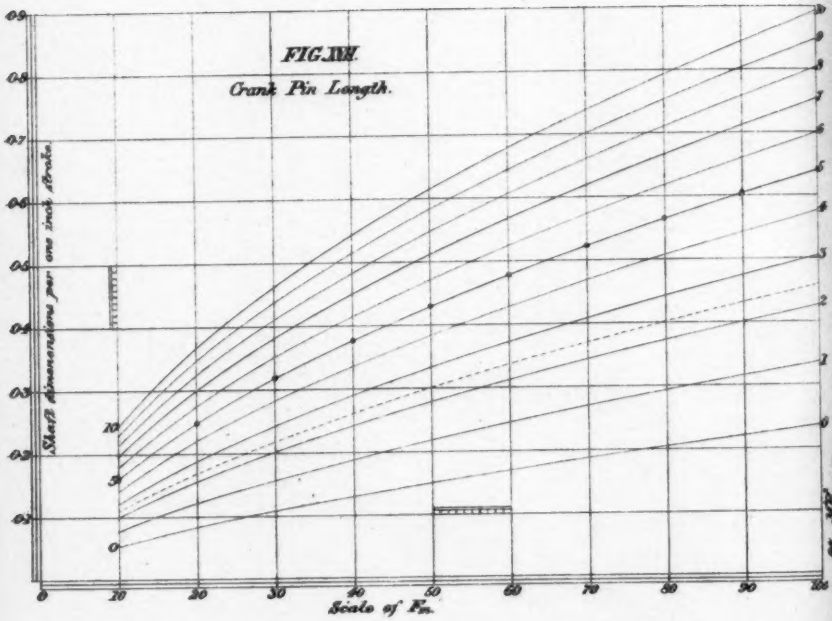
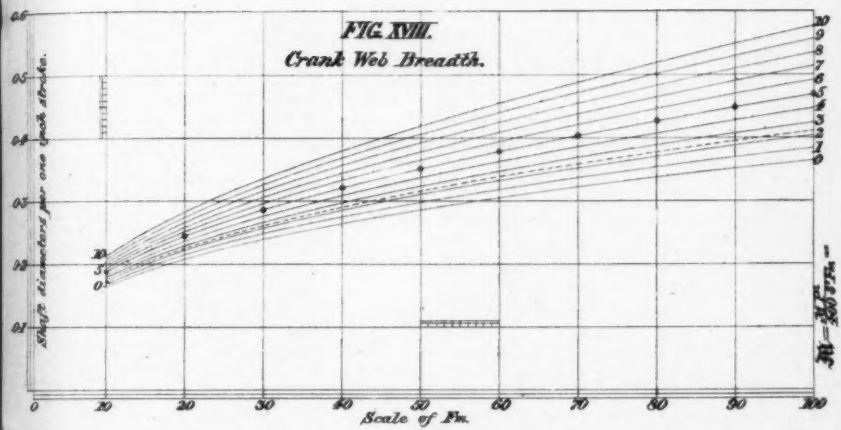


FIG. XII.
Crank Pin Length.
Shaft dimensions per one inch stroke.
Scale of Fm.
0
10
20
30
40
50
60
70
80
90
100



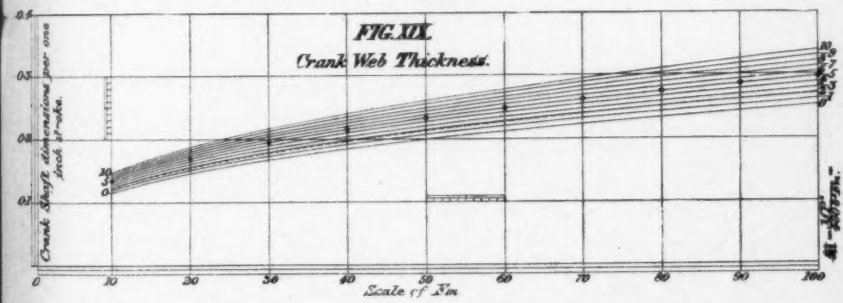


TABLE OF MULTIPLIERS.

F_m	50.	60.	70.	80.	90.	100.
10	1.006	1.015	1.023	1.030	1.036	1.040
9	1.003	1.013	1.020	1.027	1.033	1.038
8	...	1.010	1.018	1.024	1.030	1.035
7	...	1.006	1.014	1.021	1.027	1.032
6	...	1.002	1.010	1.017	1.023	1.028
5	...	...	1.006	1.013	1.019	1.024
4	...	...	1.002	1.008	1.014	1.019
3	...	...	...	1.003	1.009	1.014

Second aft bearing, *B*, removed and other bearings in line—stress in metal = 15,500 pounds per square inch.

Multiplying the ordinates of the full line curves, Figs. XVIII and XIX, by the multipliers in this table corresponding to the same values of F_m and $\mathfrak{M}$ will give the required dimensions of crank web.

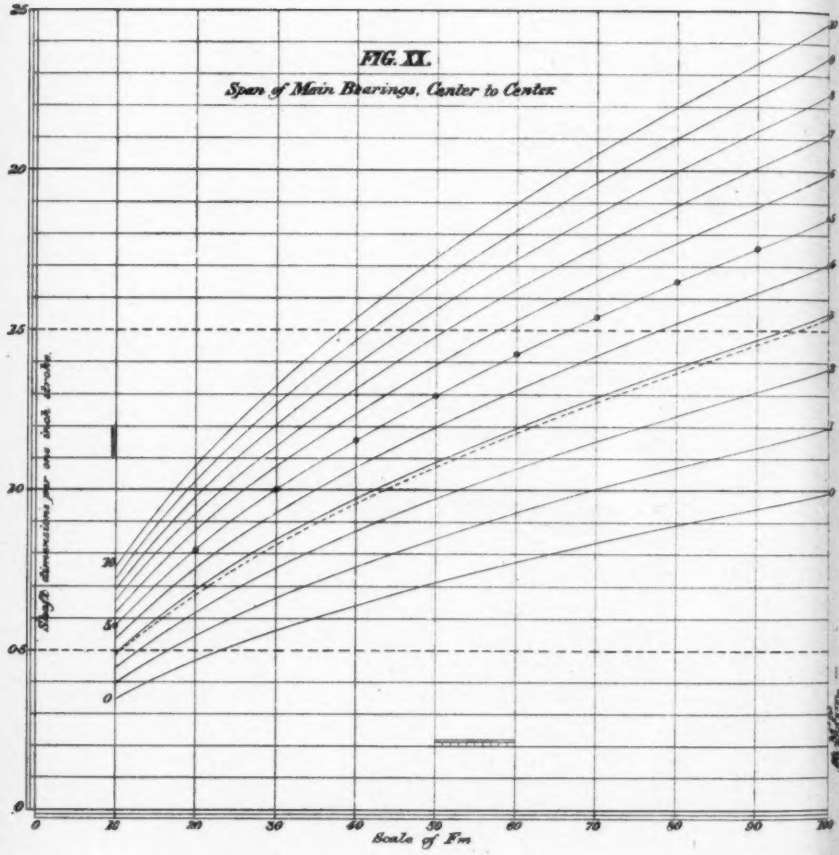


FIG. XX.
Span of Main Bearings, Center to Center
Shaft Revolutions per one Stick Stroke
Scale of Fm
0
10
20
30
40
50
60
70
80
90
100
25
20
15
10
05
0

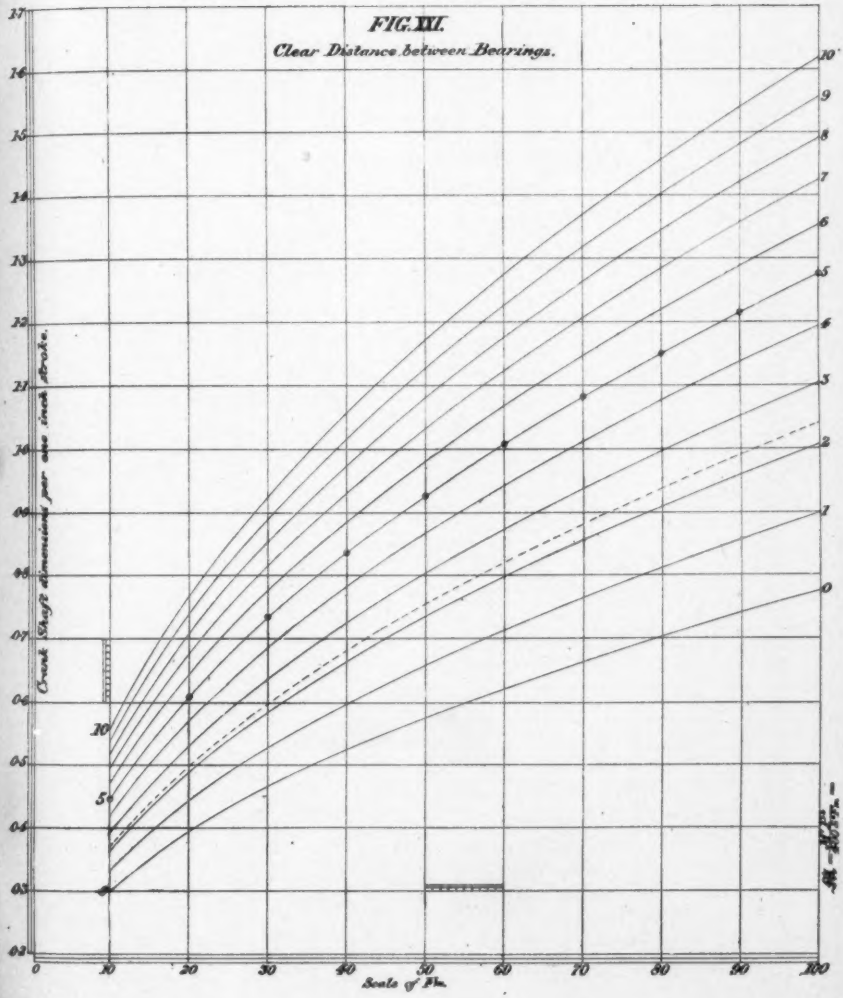


FIG. XII.
Clear Distance between Bearings.
Crank Shaft diameter per one inch stroke.
Scale of Ins.
0
10
20
30
40
50
60
70
80
90
100

HOLLOW STEEL FORGINGS.

By H. F. J. Porter, Chicago, Illinois.*

[Read before the American Society of Mechanical Engineers, 1896.]

By this name I do not refer to steel forgings that have been forged solid under a hammer or press and subsequently bored, but to forgings which have been "forged hollow."

In order that I may not be misunderstood, I will briefly describe the process of making the type of forging referred to. Supposing that we have designed a hollow shaft or roll or cylinder of outside diameter D, inside diameter D^1, length L, and solid volume $\frac{1}{4}\pi D^2 L$, an ingot must be cast solid on end to forge it from, having a diameter $2D$, volume equal to $\frac{1}{3}\pi D^2 L$, but with length L^1. From the upper part of this ingot 25 per cent. is cut off and rejected, and a hole of diameter D^1 is bored through the longitudinal axis of the remainder. The ingot is then reheated, a mandrel of diameter D^1 is run through the axial hole, and then the mass is forged down under a hydraulic press from $2D$ and length L^1 to D and L, the inside diameter D^1 remaining the same.

The reasons for adopting this method of producing hollow forgings are many and various and the result of long experimenting. In the first place, as the walls of hollow forgings are comparatively thin, the metal must be absolutely without flaw or defect of any kind, homogeneous throughout, and thoroughly worked. For this purpose, therefore, only open-hearth steel is used, and of a grade that will best insure satisfactory working. Its carbon may vary according to the purposes to which the forging will finally be applied, but its phosphorus and sulphur should not exceed .04 per cent. In order that the metal should be sufficiently worked to give it strength and toughness, the

* The Journal is indebted to the kindness of Mr. Porter for the use of the cuts. Some of the formulæ in this reprint differ from those published in the original paper, the changes having been made at the request of the author. The Journal is also indebted to Mr. Porter for the proof sheets of a similar paper on "Steel Forgings," read by him before the Western Society of Engineers, and published in the proceedings of that Society for January, 1897. Part of this second paper follows the above reprint.—Edit.

FIG. 1.—FLUID COMPRESSION PLANT.

best practice requires that the ingot should be at least twice the diameter of the finished forging. A 24-inch or 36-inch shaft or roll, for instance, would be worked down from 48-inch and 72-inch ingots respectively. Ingots of these sizes are liable to contain "blow holes," "piping" and "segregation." The upper and central portion, being the locality of last cooling, is the place where these defects are apt to occur. As a precautionary measure, therefore, ingots are made from 25 to 30 per cent. longer than necessary, so that if any defects should happen to occur, the part containing them can be cut off. Of all the various methods that have been devised to secure ingots which are solid and homogeneous and free from all of the above defects, the most efficient and most to be depended upon is the "Whitworth process of fluid compression." This consists in subjecting the metal in the mould, while fluid, to hydraulic pressure up to 7,000 tons if necessary (Fig. 1). This pressure is continued until the metal is solid throughout, great care being taken to cool the ingot slowly and equally on all sides to prevent strains or cracks forming from unequal contraction (Fig. 2). After it has

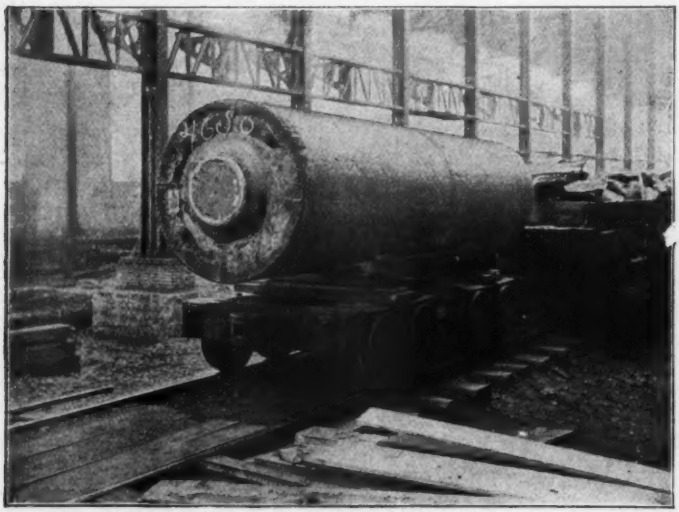


FIG. 2.—INGOT SOLID AS COMING FROM THE MOULD.

cooled, the upper part is cut off, and a hole nearly the size required in the finished forging is bored through it (Fig. 3). This cutting off of the top and boring out of the center take away those portions of the ingot where impurities may have concentrated and where there may have been a tendency toward "piping," and we now have a piece of steel which is as nearly perfect as can be produced, and it is ready for the forging process.

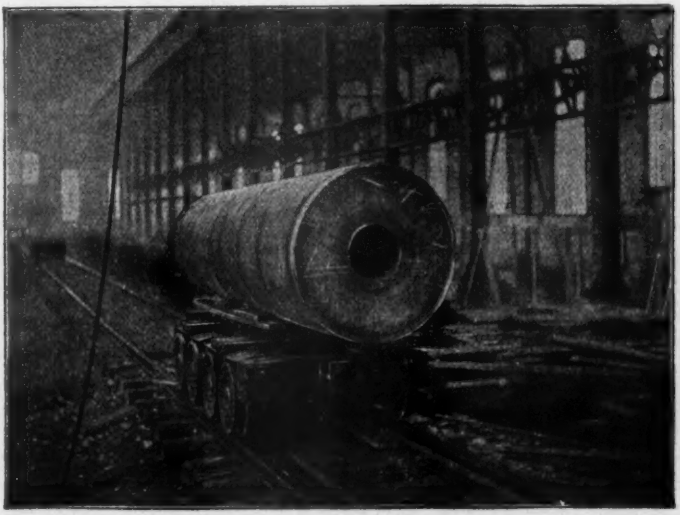


FIG. 3.—BORED INGOT.

First, it must be reheated, and as much care has to be taken in this process as was taken in its cooling, one being simply the reverse of the other. The heat must penetrate it slowly and uniformly, and its hollow shape now assists in accomplishing this result. In reheating a solid ingot, cracks are apt to form in the center, owing to the expansion of the exterior from more rapid heating. The hole in the center of the hollow forging, however, allows the interior and exterior to expand together, and relieves this tendency to crack. After being reheated, a mandrel of the proper size to fit loosely into the hole is inserted, and

the piece is taken to the press, where the metal is drawn out over the mandrel to the required dimensions (Fig. 4).

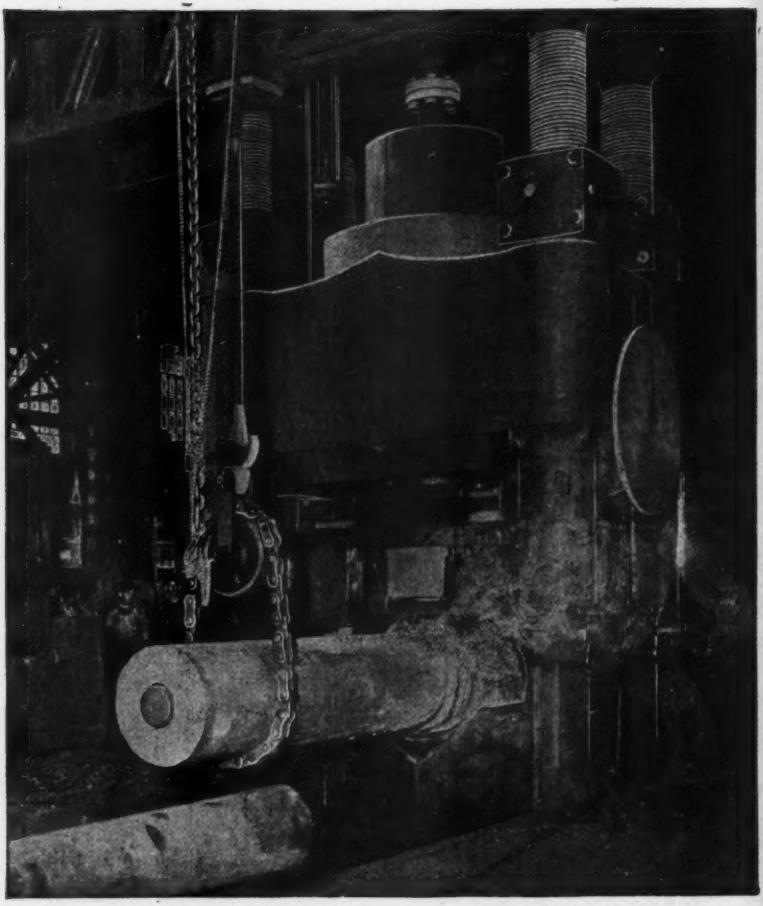


FIG. 4.—HOLLOW-FORGING A SHAFT UNDER 5,000-TON HYDRAULIC PRESS.

One of the primal requisites in forging is the proper selection of forging tools, suitable in design and power for the work in hand. The pressure applied in shaping a body of steel should be sufficient in amount and of such character as to penetrate to the center and cause flowing throughout the mass. As this flowing of the metal requires a certain amount of time, the required pressure should be maintained throughout a corresponding period. The hydraulic press instead of the hammer is therefore used to work it into shape. Under its action the forging is slowly operated upon and the pressure distributes itself evenly throughout the mass, whereas under the high velocity of impact of the hammer the metal does not have time to flow, and thus internal strains and possibly cracks are caused. The latter would be fatal defects, for, as steel has not the property of welding, they cannot be remedied. Besides, the undesirability of using the hammer on steel for the above reasons, it is a very difficult matter to make a forging of this character, except by the use of the hydraulic press. A slow and even pressure is absolutely necessary to draw out the thin cylindrical walls equally and make a forging that is straight and symmetrical throughout. The varying impact of the hammer works the metal so unevenly that the mandrel is apt to stick fast in the hole. For this type of steel forging, therefore, it is practically imperative that the press be used, and thus the metal is unavoidably subjected to the best method of treatment.

During this process of working down the metal from $2D$ to D and extending it from L^1 to L (Figure 5), it is probable that the entire piece, or at least the end of it which is being worked upon, will have to be reheated from time to time. Operating on metal which has become too cold to flow would injure it by disturbing the continuity of its structure, and thus establishing lines and planes of weakness.

Considering all the manipulation to which the piece has been subjected during the process of shaping it to the proposed design, together with its frequent partial heatings and irregular coolings, it undoubtedly has strains set up in it. To relieve these strains, it must be subjected to a final treatment of "annealing." This

treatment consists in heating the forging slowly in a furnace and then allowing the latter to cool down slowly with the forging in it. *All* forgings, whether hollow or not, should be annealed, otherwise there is a certainty of "forging strains," developing into weakness after they have been in service, causing them to get out of true, with a possibility of their breaking, particularly if subjected to alternating stresses, as in heavily weighted shafts or connecting rods, and especially piston rods, which are subjected

FIG. 5.—SHAFT IN PROCESS OF BEING FORGED.

to changing temperature. Annealing not only relieves these strains, but gives a finer grain, and increases toughness.

On this and other treatments of steel after forging depend to a great extent its physical qualities, and its strength will vary accordingly from that of the best wrought iron to between three and four times that amount. I refer not so particularly to the "ultimate strength" as I do to the "elastic limit" and toughness and ductility, as shown by the "elongation" and "contraction of area." Annealing generally lowers the elastic limit slightly

in well-made forgings, annealed forgings showing it to be about 47 per cent. of the ultimate strength. It considerably increases, however, the "elongation" and "contraction." If it is desired to develop these qualities to their fullest extent in any grade of steel, "tempering" is resorted to. This consists in heating the forging to a temperature which experience has shown to be right

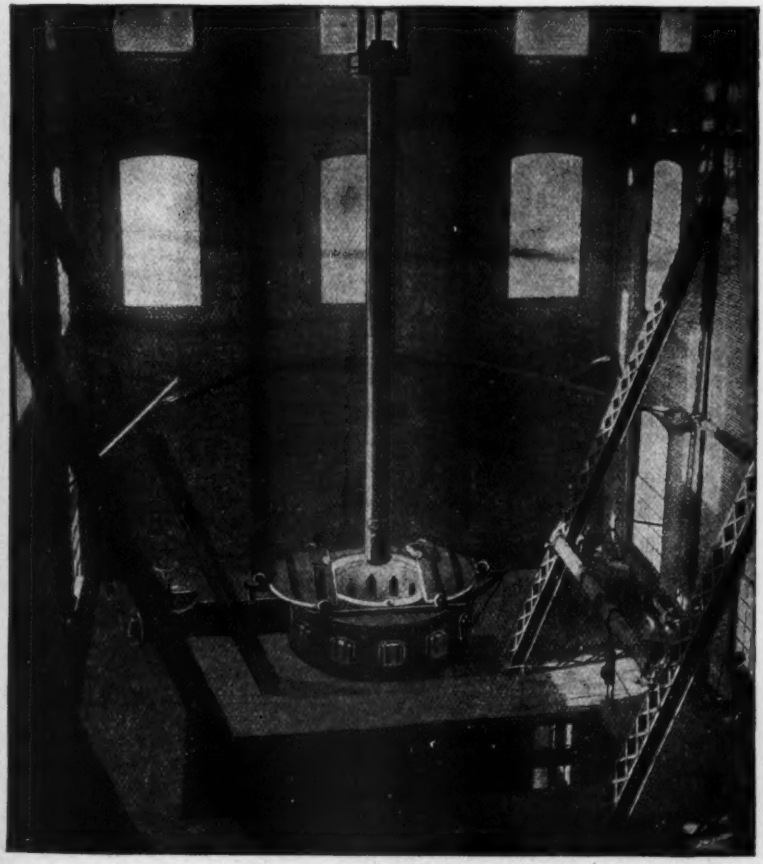


FIG. 6.—TEMPERING PLANT.

according to the purpose to which the forging is to be put, and then plunging it into a bath of oil or other suitable liquid. It is then carefully annealed. This double treatment (which is properly covered by the one word "tempering") tends to harden it, breaks up the crystalline structure due to forging, and modifies the physical properties by increasing the elastic limit and adding toughness. Forgings must be hollow to be tempered successfully, to allow the heat to be drawn from their interior as rapidly as from their exterior when they are subjected to the cooling effect of the bath, otherwise strains may be induced which result in weakening instead of strengthening them.

Authorities on machine design (*vide* Unwin, Seaton, *et al.*) say that solid shafts up to 10 inches in diameter may be subjected to a fiber strain of 9,000 pounds in wrought iron and 12,000 pounds in steel. Above 10 inches in diameter, however, iron shafts must not be subjected to more than 8,000 pounds, and steel shafts to more than 10,000 pounds. The reason assigned is, that forges do not possess hammers heavy enough to affect the center of shafts larger than 10 inches in diameter; or if by top steam or long drop they are able to be felt through the whole forging, the effect is produced by velocity of impact rather than by weight of falling mass. This, however, damages the surface, having a tendency to draw it out, and leave the central portion behind, thus producing a tearing strain on the core, causing at times actual cavities. Heavy shafts forged under light hammers show the effect of this treatment in having concave ends (Fig. 7), while shafts forged under sufficiently heavy hammers or under the hydraulic press show bulging ends, the metal in

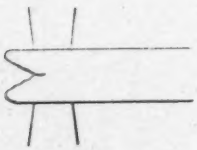


FIG. 7.

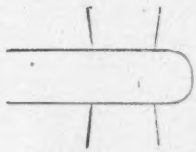


FIG. 8.

the center, where it is hottest and softest, being pressed out (Fig. 8). Another reason why authorities place a higher safety

factor on large shafts, especially of steel, is on account of the liability of steel ingots to have "piping" and other defects in their center.

With hollow forgings manufactured under the processes here described, these objections are met by using fluid compressed steel and subsequently taking out the possibly defective center altogether. By substituting for the center, during the process of forging, a mandrel, the latter acts as an internal anvil, and by this means even in the largest hollow shafts the thickness of metal worked upon can be brought within the limits above mentioned. The use of hydraulic presses having a capacity of from 2,000 to 14,000 tons, selected according to the size of the forging under treatment, can leave no doubt in the most incredulous minds that the metal has been thoroughly worked. Take, for instance, the Ferris wheel shaft, the largest ever made, 32 inches in diameter, with a 16-inch hole through it (Figure 9). The walls of this shaft are only 8 inches thick between the mandrel and the press, and

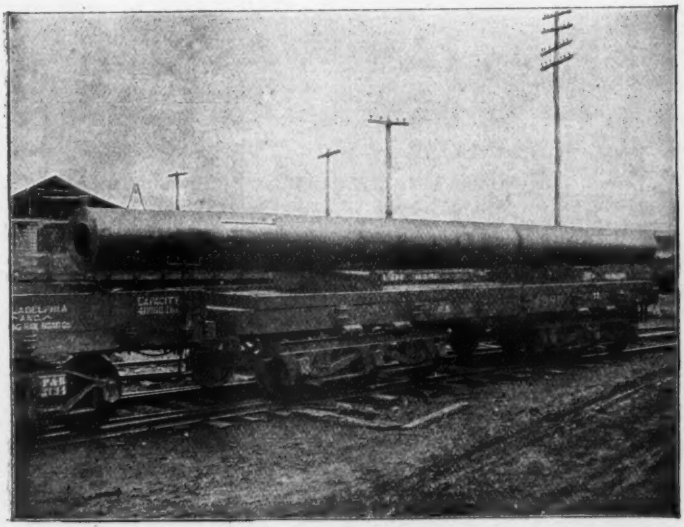


FIG. 9.—THE FERRIS WHEEL SHAFT.

the metal, therefore, is in better condition, as far as its physical properties are concerned, than in an 8 inch solid shaft, the center or poorer portion of which has not been removed.

By the use of the usual formulæ, we have a means of comparing figures. For solid shafts of diameter d in inches:

(1) When the shaft is subject to *twisting* only,

Let P = twisting force in pounds at end of lever arm.

r = lever arm in inches.

s = safe shearing resistance of the metal in pounds per square inch.

Twisting moment $T = Pr$ = moment of resistance $= \frac{\pi d^3 s}{16}$, and

$$Pr = .1963 d^3 s, \text{ and } d = \sqrt[3]{\frac{5.1\ Tr}{s}}.$$

(2) When the shaft is subject to *bending* only,

Let P = bending force in pounds.

l = distance between supports in inches.

t = safe tensile resistance of the metal in pounds per square inch.

Bending moment $B = \frac{1}{4} Pl$ = moment of resistance $= \frac{\pi d^3 t}{32}$, and

$$Pl = .3927 d^3 t, \text{ and } d = \sqrt[3]{\frac{2.55\ Pl}{t}}.$$

And for hollow shafts, letting d and d' = the outside and inside diameters in inches, as $d^3 = \frac{(d^4 - d'^4)}{d}$, by substituting in the above we have,

(3) $Pr = .1963 \frac{(d^4 - d'^4)s}{d}$. $\quad d = \sqrt[3]{\frac{5.1\ Tr}{\left(1 - \frac{d'^4}{d^4}\right)s}}$.

(4) $Pl = .3927 \frac{(d^4 - d'^4)t}{d}$. $\quad d = \sqrt[3]{\frac{2.55\ Pl}{\left(1 - \frac{d'^4}{d^4}\right)t}}$.

(5) When the shaft is subjected to both twisting and bending simultaneously, the combined strain may be measured by calculating what is called the equivalent twisting moment T'.

If T = the twisting moment and B = the bending moment as above, the equivalent twisting moment

$$T' = B + \sqrt{B^2 + T^2}.$$

Assuming that steel less than 10 inches in thickness may be submitted to a fiber strain of 12,000 pounds per square inch, and when 10 inches thick and thicker it must not be submitted to a greater strain than 10,000 pounds per square inch, we may so reduce the thickness of metal operated upon in shafts of larger diameter than 10 inches, by hollow-forging them, that the walls will be less than 10 inches in thickness.

By the above formulas, hollow-forged shafts are shown to be as strong or stronger than solid shafts of the same diameter. Taking for example the shaft above mentioned, 32 inches outside diameter, 16 inches inside diameter, we find that when compared with a solid shaft of the same size it has lost 25 per cent. in weight and gained 8 per cent. in strength.

With the substitution of steel for wrought iron in the trades for engine and miscellaneous forgings, the tendency has naturally been to use a mild or soft steel approaching iron as regards physical qualities and in the ease with which it can be machined. Wrought iron has a low elastic limit, averaging about 20,000 pounds per square inch in large sections, where proper care is taken in its production. Its strength is apt to be impaired by imperfect welds and porous spots enclosing slag and scale.*

Although mild steel, when of good quality, is superior to wrought iron in strength, toughnesss, homogeneity, and freedom from danger of such defects, still it does not possess the very desirable quality of high elastic strength combined with ductility or toughness in as great a degree as can be obtained without danger in a harder steel, when proper precautions are taken in its manufacture. In other words, in the use of ordinary mild steel, only a partial advantage is taken of the most desirable qualities of steel which are easily within reach. In some instances where the amount of machine work in finishing is very great and there is ample margin of safety in the design—as, for

*See extract at the end of this paper.—EDIT.

instance, is often the case with connecting rods—the use of mild steel may be advisable. Such steel contains about .20 to .25 of one per cent. carbon, and can be guaranteed to show, in specimens $\frac{1}{2}$ inch diameter and 2 inches long between measuring points, cut from full-sized prolongations of forgings or from representative pieces, a tensile strength of not less than 57,000 pounds per square inch, and an elastic limit of not less than 27,000 pounds per square inch, with an average elongation of 25 per cent.

For the general run of engine forgings, however, a harder steel should be used in which a tensile strength of about 75,000 pounds and an elastic limit of 35,000 pounds per square inch can be obtained, together with an average elongation of 20 per cent.

When proper precautions are employed, forgings can be made with perfect safety of a still higher grade of steel, and this is especially recommended for crank and crosshead pins and for machine parts subject to severe alternating strains and wearing action. In this grade of steel, a tensile strength of about 85,000 pounds and an elastic limit of about 40,000 pounds per square inch can be obtained, with an elongation of 15 per cent.

If steel forgings are tempered, they will possess still higher qualities than those above mentioned, and can be furnished with a tensile strength of 85,000 to 90,000 pounds and an elastic limit of 45,000 to 55,000 pounds per square inch and an elongation of 20 to 15 per cent.

By introducing about 3 per cent. of nickel into the composition of steel, a finely granular or amorphous condition is obtained in forgings, and the very highest quality of steel is attained.

By the combination of hollow-forging and tempering this nickel steel, a material is obtained excelling all others known in elastic strength and toughness. As an example of this can be mentioned shafts made for the United States warship *Brooklyn*. These showed in specimens cut from full-sized prolongations a tensile strength of 94,245 pounds, an elastic limit of 60,775 pounds per square inch, an elongation of 25.55 per cent. and a contraction of area of 60.58 per cent.

Professor Merriman is quoted, in a paper read before the Society of Naval Architects and Marine Engineers in 1893 by R. W. Davenport, as estimating the strength of these shafts compared to solid shafts as follows, when strained to one-half their elastic limit:

1. Propeller shaft United States steamship *Brooklyn*, nickel steel, hollow forged; outside diameter, 17 inches; inside diameter, 11 inches; weight, 19,112 pounds.

a. Horse powers transmitted at 50 revolutions per minute, 15,780.

b. Load in pounds at middle of a span of 12 feet on two supports, 276,200.

2. Simple steel shaft, solid, 13 inches diameter (same weight and sectional area as above).

a. Horse powers transmitted under similar conditions, 5,130.

b. Load in pounds under similar conditions, 89,000.

Comparative strength of these two shafts as 3 to 1.

3. A solid shaft of simple steel of the same strength as the hollow-forged nickel steel shaft would be 18.9 inches diameter, and weigh 53 per cent. more.

Hollow-forged fluid-compressed steel of .40 to .45 of one per cent. carbon, and more especially nickel steel, oil-tempered, is markedly adapted for piston rods of rock drills, mining machines, and hydraulic presses, and for drop-hammer rods, stamp stems, cam shafts, and similar pieces that are subjected to stress alternating between tension and compression, or of either kind, frequently repeated. By substituting steel of this grade, which would have an elastic limit of about 60,000 pounds per square inch, for wrought iron or mild steel, which is generally used for the purpose, and by so proportioning the cross section that the metal is not strained beyond one-half the elastic limit, so-called "crystallization" from shock or vibration does not occur and its life is prolonged indefinitely.

Undoubtedly the best type of hollow forging, and one which is gradually being adopted both for shafts and rolls, is where the walls are of the same thickness throughout, the outside and inside diameter varying together, both being greatest at the center,

where the strength is required, and smallest at the bearings (Fig. 10). Such a shaft is built on the principle of a girder, and offers the greatest strength for the least amount of metal.

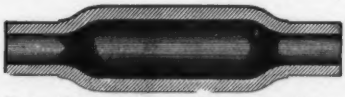


FIG. 10.—A HOLLOW SHAFT OR ROLL.

The ability to produce forgings of this hollow variety has led to their adoption in many places where castings, both of iron and steel, have previously been used. This substitution has resulted in considerably lightening the dimensions of such pieces, and also the parts in which they rest or move.

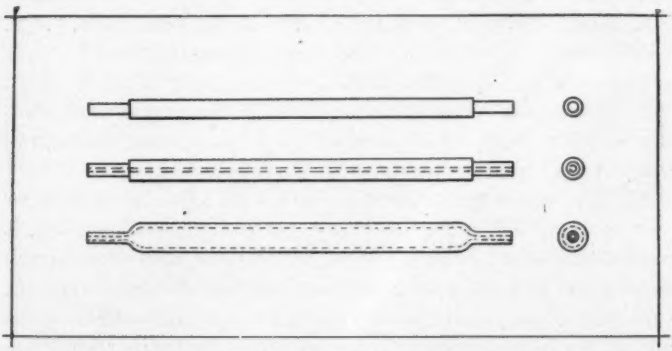


FIG. 11.

Prominent among hollow forgings are the Ferris wheel shaft, 32 inches outside diameter, 16 inches inside diameter, 45 feet long, weighing 89,320 pounds; and shafts for the steamers *Puritan*, *Plymouth* and *Pilgrim* of the Old Colony Steamboat Company, 20½ inches outside diameter, 9 inches inside diameter, and weighing 65,832 pounds each. Shafts for the Navy are made hollow, and those for the steamers *New York*, *Paris*, *St. Louis* and *St. Paul*, of the International Line, are 21 inches in diameter, crank pins 22 inches in diameter, with a 10-inch hole through both. Field rings of nickel steel were hollow-forged for the 5,000 horse

power Westinghouse dynamos at Niagara Falls, 11 feet $7\frac{3}{8}$ inches diameter, $4\frac{11}{16}$ inches thick and 4 feet $2\frac{3}{4}$ inches wide, weighing 28,840 pounds (Figure 12).

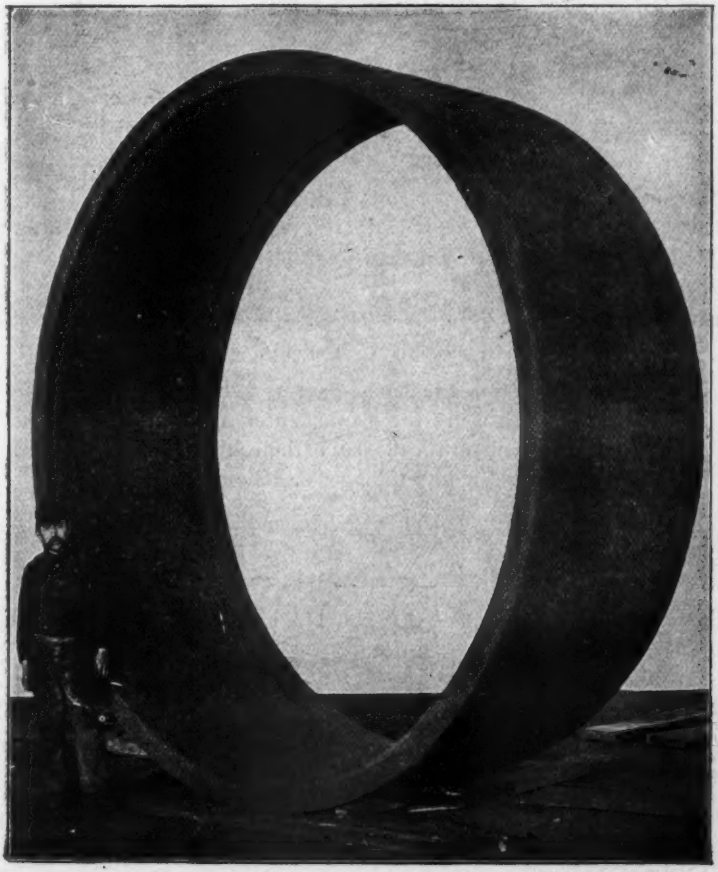


FIG. 12.—FIELD RING FOR 5,000 HORSE POWER DYNAMO.

Thus it will be seen that there is scarcely any limit, either in length or diameter, to the proportions of forgings which can now be made by processes which have been perfected. In this, as in

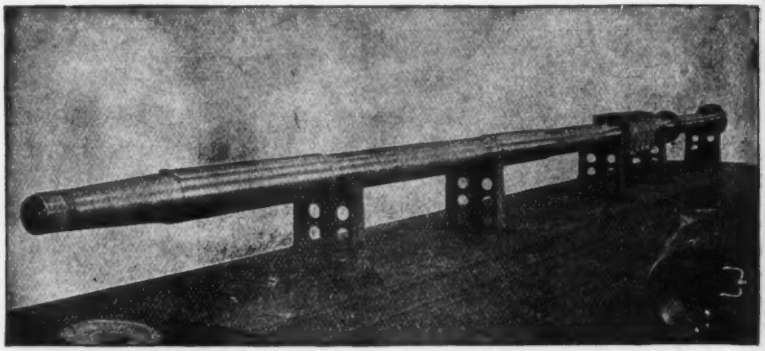


FIG. 13.—PROPELLER SHAFT FOR STEAMERS "ST. LOUIS" AND "ST. PAUL:" LENGTH, 53 FT. 5 IN.

every other direction, the demand seems to have been more than met by the enterprise of the manufacturer, and as a wide field is opened before them for the display of their prowess, it would seem as if engineers are rapidly taking possession of it.

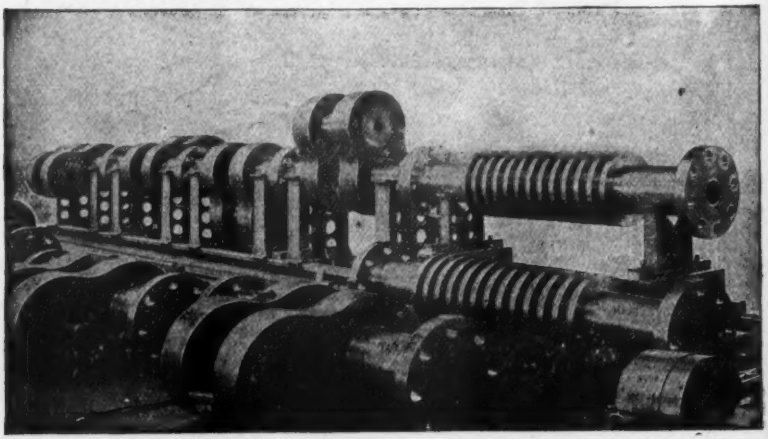


FIG. 14.—CRANK AND THRUST SHAFTS FOR STEAMERS "ST. LOUIS" AND "ST. PAUL."

STEEL FORGINGS.

By H. F. J. Porter, Chicago, Illinois.

[Extract from a paper read before the Western Society of Engineers, Chicago, 1896.]

When the Bethlehem Iron Company built their forge they decided that they would turn out steel forgings only. There were various reasons for coming to this determination, but the principal one was that they had already passed through the stage where they had changed the product of their rolling mill from iron to steel, and they knew the benefit that would accrue to forgings by making them of steel also. I could go into various other reasons which determined them to adhere to steel and not take up iron, but I cannot give you some of these more readily than by reading a few words from what W. F. Durfee, of New York, has to say on the subject of "Wrought Iron." The article from which I quote appeared in the August number of the Iron Age and is a copy of a lecture delivered before the Franklin Institute a short time previously.

"The term 'wrought iron' is popularly supposed to designate a metal, but it is really the name of a mechanical admixture which, at its best, consists of clusters of crystals (which may with propriety be regarded as compound crystals) of practically pure iron, separated from one another, as the result of the manipulative processes employed, by films or threads of an unavoidable impurity called 'cinder.' * * * In the manufacture of wrought iron, the pig, or other variety of cast iron, is first deprived in a more or less imperfect degree, of its carbon and other impurities by what is known as the 'puddling process.' This process may be briefly described as consisting of four distinct operations, viz.:

"1. The melting of the pig iron.

"2. The boiling of the melted metal in a bath of liquid cinder (composed mainly of silicate of protoxide of iron) until the iron (which owing to its loss of carbon and other impurities can no

longer remain fluid at the temperature employed) begins to solidify in the form of small granules or crystals, which can be seen moving amid the boiling cinder like white hot peas in a red hot soup. When the iron begins thus to granulate or crystallize, it is said to be 'coming to nature.'

"3. The collection, by the puddler, of these granules or crystals into distinct masses called 'balls' which may with propriety be regarded as white hot sponges of iron saturated with liquid cinder, which fills all their numerous accidental and irregular cavities.

"4. The squeezing or hammering of these balls while still at a welding heat into more solid masses, which are called blooms. These contain much less cinder and other impurities than the balls, but are far from being uniform in structure; for when the balls are squeezed or hammered (this last operation is often called 'shingling') for the purpose of expelling the cinder and welding the granules or crystals of iron into a homogeneous mass, the attempt is never wholly successful; also as the metal cools, the cinder quickly acquires a pasty consistency and flows with difficulty, and a large portion inclosed in the interior cavities of the ball is merely flattened or elongated. Hence, it will be seen that the bloom is composed of a compact mass of granules or crystals of iron, separated from one another at numerous points by films, layers or strings of cinder of very irregular dimensions. * * * When a properly heated bloom or other similarly constituted mass of wrought iron is subjected to the action of the hammer or rolls, the contained cinder endeavors to escape from its entangled mechanical alliance with the crystals of the iron, and in so doing each particle thereof is driven into some line of least resistance, which is always finally located in a plane at right angles to the direction of the force acting upon the metal. * * * The direct consequence of the elongation of its compound crystals, and the effort of the intervening cinder to escape in the direction of least resistance while the wrought iron bloom is being forged or rolled, as before described, is the establishment of that structural peculiarity in the resulting bar known as 'fiber,' which is one of the most conspicuous features of wrought iron and one not found in any other variety of ferruginous materials. When certain of the

films or threads of cinder in a bar of wrought iron are so large as to be distinctly visible on its surface to the unassisted eye, they are called 'sand seams' or 'cinder cracks.'

"* * * Steel is iron freed from mechanically mixed impurities, such as cinder, etc., by a melting process during which there is combined with it chemically a small percentage (not large enough to prevent the metal being forged or rolled) of other impurities, introduced for the purpose of modifying its strength, hardness, elasticity or ductility in such a way and degree as to adapt it to the particular use to which it is to be applied. In short, while wrought iron is iron having (as the unavoidable result of the methods employed in its manufacture) its impurities mechanically mixed therewith, steel is iron having (as the result of the adoption of appropriate manufacturing processes) its impurities chemically combined."

I have gone rather at length into this paper, because Mr. Durfee has put in a few words a great deal of food for reflection. Many people say that they prefer iron to steel even to this day on account of "fiber." If they would only look into the subject a little, they would find that "fiber" is nothing but cinder which cannot be a desirable constituent of a forging which must depend upon the soundness of its welds to hold it together. Mr. Durfee makes that very plain. What is wanted is a metal free from impurities so that molecular attraction can be depended upon to give it strength. Such metal is obtained by the process adopted in the manufacture of steel. In that process the metal is melted and the cinders float on top and are taken off, leaving a homogeneous mass underneath, and the only impurities that are in it are what have been chemically combined in it for the purpose of increasing its strength or modifying its physical properties.

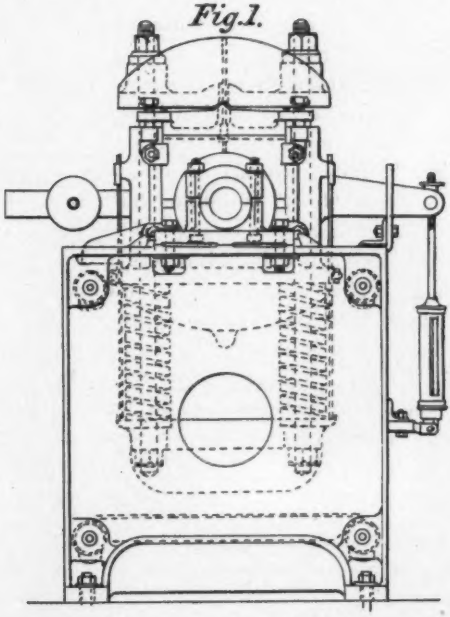
MACHINERY BEARINGS.

By John Dewrance, Assoc. M. Inst. C.E.

[Read before the Institute of Civil Engineers, 1896.]

In this paper are presented the results of a series of experiments undertaken by the Author to determine the frictional resistance to shafts revolving in bearings under varying loads, when subjected to different conditions. The testing machine

Fig.1.



used is shown in Figs. 1. The journals upon which the experiments were made were 10 inches and 4 inches respectively in diameter, and formed part of a shaft supported at its ends in

plummer blocks. A saddle rests upon the upper bearing and is connected through four bolts compressing springs to a plate pressing upon the under side of the lower bearing. The load carried by the journal can therefore be varied with the degree of compression of the springs, the pressure upon the upper bearing being greater than that on the lower by an amount equal to the total weight of the saddle, spring and bearings, generally about 16 cwt. It was not convenient to apply a thermometer to the

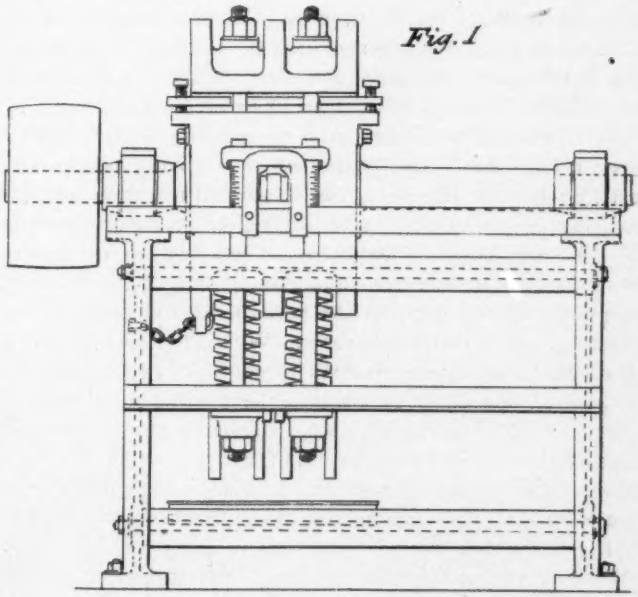


Fig. 1

lower bearing, so the observations were usually confined to the upper one, which, having to support the greatest load and being at a disadvantage as regards lubrication, generally failed first When the saddle had been fitted with a pair of bearings the four bolts were evenly screwed up, the amount of compression being indicated by scales and pointers on each side. When loaded, the saddle was free to turn with the shaft through a certain range,

and a spring balance, fitted, as shown in the Figure, to retain the saddle vertical, indicated the amount of the force required to turn the shaft.

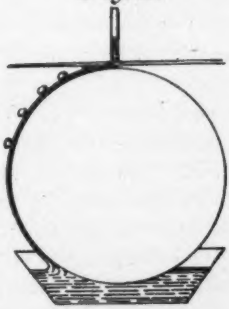
The first experiments were made upon a pair of 10-inch by 16-inch bearings having two oil holes on the top center line. They were carefully fitted about $\frac{1}{64}$ inch larger in diameter than the journal, but they would not run cool with the weight of the saddle alone. When a channel was made to connect the two holes and to lubricate the part of the bearing between them, the result was not greatly improved. When oil was thrown upon the shaft at the openings between the bearings they began at once to sustain loads more or less satisfactorily, but the oil issued from the holes at the top. The pressure indicated on a gauge connected to the center of the bearing was equal to the greatest load that could be applied to the large bearing ; the experiments were, therefore, transferred to the 4-inch shaft with a bearing upon which the same load represented a greater pressure. A long series of experiments resulted ultimately in a pressure of 2,300 lbs. per square inch being recorded. The bearing was taken out and the surface dressed with a view to obtaining even greater pressures. When, however, it was replaced the following day, instead of continuing to deliver oil at the gauge connection as hitherto, the bearing would take oil rapidly. The pressure gauge was removed and a vacuum gauge substituted, when it was found that a vacuum equivalent to 30 inches of mercury existed where previously a high pressure had been recorded. The bearing still ran cool and well, as shown in Table I of the Appendix.

When taken out there was nothing in its appearance that at first sight accounted for the change described, but further examination revealed that, although the back of the bearing had been planed, it was, from some cause, not quite flat, the center of the bearing taking all the load. This had sprung the bearing slightly and had worn the central part of the surface. When the load was removed the center sprang back, leaving that part of the surface round the hole separated from the shaft. The load applied during subsequent experiments had not been suffi-

cient to spring the bearing flat again. The experiments on this point were continued until a bearing was produced that, when placed on to the shaft one way, gave a pressure in a hole at the center of the bearing and, when reversed, gave a vacuum at the same point. As a result of the observations made during the foregoing experiments, an arrangement was set up which is shown diagrammatically by Fig. 3 to illustrate the way in which oil lubricates a bearing.

If oil is slowly poured on to a revolving shaft, it accumulates in a thick ring, most of which can be wiped off by the hand. The oil next to the shaft adheres more firmly than that which is further away. If the oil ring be imagined to be composed of films of oil, each of the thickness of a molecule of oil, the first film has the solid shaft to adhere to. The second is separated from the shaft, but it can adhere to the first, which is itself firmly held. Each successive film is less influenced by the force of adhesion that exists between the shaft and the oil, until the point is reached at which centrifugal force is stronger than the adhesion and the oil flies off the shaft. If a shaft be rotated in a bath of oil, the ring of oil is also formed. In Fig. 2 the circle represents a shaft,

Fig. 2.



on the top of which a flat plate rests. However light this plate may be, it affects the ring of oil immediately it touches it. Unless the plate entirely stops all the oil from passing round with the shaft it must float upon that part of it which remains between

it and the shaft. There must, in other words, be a film of oil at a sufficient pressure per square inch acting on the narrow surface of contact to support the total load put upon the plate. This pressure is the result of the multiplication of the force of adhesion of the oil to the shaft by the inclined plane formed by the plate to the shafts.* The velocity of the oil is also increased in the same way. If the load on the plate is a light one the adhesive force is sufficient, when increased by the inclined plane, to produce the pressure required to lift the plate and carry several films round with the shaft; but, as the load is increased, fewer films are able to pass the plate until the point is reached at which the surfaces abrade one another for want of lubrication.

If a hole is made through the plate, this pressure can be observed, increasing as the hole approaches the point at which the plate rests upon the shaft. If the hole is pushed beyond this point a vacuum is produced. The oil and air in the hole adhere to the films and are carried round with the shaft by surface adhesion. A flat plate was employed in this experiment, as it was easier to observe the relative position of the hole. The conditions are not greatly changed if the diameter of the shaft be reduced and the plate curved. A bearing that exactly fits the shaft all the way round will not run cool, and the well known fact that a bearing must not be tight at the side indicates that the inclined plane must exist in some form or other. All successful bearings are constructed with an inclined plane in some form, and the load that a bearing will sustain is determined by the inclination. If the angle is sharp it will not multiply the adhesion of the oil so much as if it is more gentle. By suitably adjusting the inclination it has been found possible to pump oil between the surfaces to a pressure of as much as 3,000 lbs. per square inch.

These experiments were originally instituted to demonstrate whether with similar lubrication and conditions a bearing surface composed of one alloy would allow of a greater load than a bearing surface of another alloy, and they have proved that it

* Another explanation of the phenomenon is given: see "On the Conversion of Heat into Work," by Dr. William Anderson, F.R.S., 2d edition, p. 19.—SEC. INST. C.E.

will not. The composition of the metal of the bearing surface has little or no influence on the load that the bearing will support. Bearings composed of a metal that may, under certain circumstances, seize at a pressure of 20 lbs. per square inch, have been run with a load of more than 3,000 lbs. per square inch, and many different kinds of metal have been used in bearings that have run loaded to more than 1 ton per square inch. At these loads the lubrication becomes so uncertain and difficult that the point at which it fails is due to conditions that can not be observed. In no case was there the slightest evidence that this point was reached sooner with one alloy than it was with another, so long as the metal itself would support the load. The experiments leave no doubt that so long as a bearing runs fairly cool the surfaces of the shaft and bearing are separated by films of oil. If the number of films is small it is possible to have considerable heating without actual seizing, but if the films of oil are entirely absent the surfaces adhere or seize at once. The simplest example of this kind of adhesion is afforded by the abrasion of an iron surface by a piece of brass. The crystals of the iron tear out crystals from the brass. By burnishing the iron surface this tendency is reduced to a minimum, and by corroding the surface chemically it is increased to a maximum. If the pressure and speed are low, a great deal of brass can be torn from the high places of a bearing, especially if there is a good supply of oil.

When a bearing that has worn to a surface is allowed to rest upon the shaft without the intervention of a film of oil, and the crystals of the bearing adhere to the shaft, they must make an elevation on the shaft that would prevent it being turned except by a force sufficient to lift the load on the bearing to the height of the elevation due to the crystals. This would concentrate the whole load on this elevation, with the result that more crystals would be dragged out. If the load is very great, say more than 1 ton to the inch on an 8-inch by 4-inch bearing, seizure has occurred with a suddenness almost startling; but the presence of oil in some parts of the bearing, and the lower loads used in practice, generally make the seizing more gradual.

A piece of iron will not leave a mark upon a surface softer than itself; it becomes coated with the softer metal. If the bearing is of a material of which the crystals are individually stronger than those of the shaft, the crystals of the shaft adhere to the bearing, which, being stationary, causes the crystals to be heaped up in one place instead of being carried round and spread over the whole circumference of the shaft. This is the reason that the seizing of a cast iron bearing is often attended with such disastrous results, and there can be no doubt that the softer the metal of the bearing the safer is the shaft from injury from seizing. Many hundreds of experiments were made upon the machine with soft metal bearings without injury to the shaft, but when similar experiments were attempted with hard bronze bearings the shaft was several times injured and had to be turned.

It is possible under some circumstances to provide sufficient lubrication without intermission. When this is the case, the shaft revolves in oil, and it is surprising with what a small power a heavy load can be supported, and how small the destruction of the surfaces may be under these conditions, as shown in Table II of the Appendix.

It is well known that even when the lubrication has been continuous the surface of the bearing has suffered considerably, and in some cases the surface of the shaft. If it is accepted that, so long as a bearing works cool and shows no sign of seizing, the surface and shaft are separated by films of oil, it is at first sight difficult to see how either metallic surface can wear away. In some cases the oil contains grit which is carried between the surfaces and scratches them. It has been proved on the testing machines that dust that will float in a quiet atmosphere is usually less in bulk than the thickness of the film of oil. Had this not been the case the experiments could not have been conducted where they were, as the machine was exposed to a considerable amount of floating dust.

The corrosive effect of the oil itself on the sufaces does not appear to have been hitherto recognized. It was first observed when experimenting with a pair of bearings of pure lead upon the 10-inch shaft. Olive oil was used, but after passing through the

bearing several times, it became black and thick. This oil, after filtration, was composed of 16 per cent. of oleate of lead, 9.57 per cent. of oleic acid and 74.62 per cent. of olive oil and glycerine. Oil of the same quality was then run through bearings composed of hardened tin, which were found to be but little affected. Disks of the metals used in the manufacture of bearings were immersed in oleic acid, and occasionally drawn up out of the acid so as to be exposed to the air. Lead and zinc rapidly corroded away; copper was corroded, but to a less extent. Tin and antimony were not appreciably affected. Oleic acid appears to attack lead, zinc and copper with great avidity. Even if the oil is free from acid it becomes charged with oxygen from the atmosphere which oxidizes the surfaces, the oxide itself being immediately carried away by the oil. A great number of experiments showed that a bearing composed of an oxidizable metal, such as hardened lead, could be worn and scraped to a surface corresponding to the shaft in a quarter of the time required to produce the same effect on a bearing composed of hardened tin. For this reason a number of the special forms of bearing was made of hardened lead. Hardened zinc was tried in one instance, the bearing being tested by hydraulic pressure after the usual pressure gauge holes were drilled, and oleic acid being used as a lubricant. The acid attacked the surface so rapidly that instead of improving it became worse as time elapsed. Oil was then used, but when the surface had arrived at the point of delivering it at pressure, it was found that the oleic acid had soaked into the pores of the metal and so corroded it that the oil oozed out all over the bearing at very slight pressure. The alloys of zinc are probably the most crystalline used for bearings, and there seems to be no doubt that the size of the crystals greatly affects the rapidity of chemical corrosion.

This open grain or crystalline structure occurs more or less in all bronze castings, and renders them more subject to chemical corrosion than would otherwise be the case with an alloy of copper and tin. The chief recommendations of bronze as a material for bearings are its high melting point, and its capacity of resisting high compressions. The melting point of the tin alloys

is 500 degrees Fahrenheit. The alloys of lead and of zinc vary more than the tin alloys, but their melting points are not much higher.

With suitable lubrication bearings should run cool; if the temperature rises above 200 degrees, the viscosity of the oil is so much reduced that the bearing will probably seize. There is considerable difference of opinion as to whether a bronze bearing will behave better than a tin bearing under such circumstances. The bronze bearing will, if allowed to run after it becomes heated, almost certainly injure the shaft, but the tin-alloy bearing will run till the temperature reaches 500 degrees, without injury to the shaft. Higher temperatures than 500 degrees are dangerous, and with proper arrangements ought never to occur.

The compression test deserves more consideration than it has hitherto received. Many of the alloys of tin and lead now used to line bearings have so low a compressibility that they yield under the ordinary pressures applied to bearings, with the result that the metal squeezes into the oil inlets and stops lubrication. This circumstance is no doubt responsible for much of the trouble that has been experienced with the use of alloys of this class. It is suggested that no alloy should be used until it has been demonstrated satisfactorily that its point of first yield is considerably above the greatest load or shock to which it will be subjected in use.

The method of making such a test is very simple. A bush 4 inches in diameter by $3\frac{1}{8}$ inches bore, giving an area of metal of 5 square inches, is cast on a chill and is placed in a hydraulic press. A line is drawn on the side with a pair of compasses set to about 3 inches radius. The bush is loaded by successive increments of $\frac{1}{4}$ ton, the load being taken off each time. When it is found that the line drawn by the compasses thickens the previous line, the metal has yielded. Even after this point, different alloys behave very differently, some taking a large increase of load to cause a yield of $\frac{1}{8}$ inch, others continue to yield very fast after they first start. It is possible to make an alloy of tin that will not yield in this way until loaded to 8 tons per square inch.

The Author's experiments suggest the following rule: The oil

should be introduced into a bearing at the point that has to support the least load, and an escape should not be provided for it at the part that has to bear the greatest load.

All the most important bearings belong to one of three classes—(A) those having a continuous load in one direction, (B) those having an alternating load in opposite directions, and (C), those with both a continuous load in one direction and an alternating load in opposite directions.

In class A is included the ordinary mill bearing or plummer block used for supporting shafting. The oil is fed into the center of the top bearing at the point that has to bear the least load, so that in this case the rule is conformed with. This class also includes railway carriage bearings. These were originally lubricated by holes through the crown of the bearing at the point of greatest pressure. It was, however, found that the lubricant would not enter at that point until the surfaces were more or less roughened. These bearings are now invariably lubricated according to the rule given. Footsteps of vertical shafts and thrust blocks of marine engines belong to this class, but no experiments were made upon them.

In class B are included the bearings of vertical engines. The bearings of a marine engine resemble the ordinary plummer block, and in the shaft tunnel their duty is the same; but those near the connecting rod have an entirely different duty to perform, which is the same as the connecting rod bearing. The oil is generally applied at the center of the top bearing in defiance of the rule. If the oil hole is left plain, it is found that no oil will enter during that part of the stroke when there is pressure on the top half of the bearing. To meet this difficulty oil channels are cut. It must be evident that as the pressure of oil between the surfaces gradually increases from the point of least pressure to a pressure equal to the load at the point of greatest pressure, channels that run circumferentially around the shaft must be bad unless they are confined to the part of least pressure. Their effect is to scrape off the oil at the point of greatest pressure, and deliver it unused at a point of less pressure. The result in the ordinary marine bearing is that the oil delivered into the bearing runs down the channels as far as

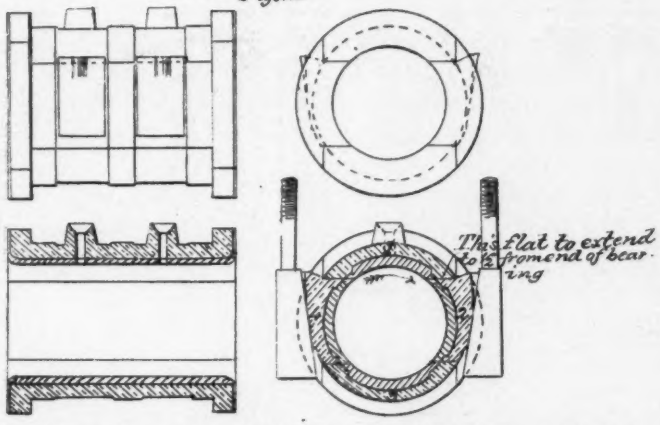
possible, and is not used on what might be called its first journey through the bearing. It is then taken up by the shaft and is carried to the second half of the bearing. The proper point to introduce the oil is just above the joint of the bearing at the side. There the oil is distributed over the shaft and carried to the point of greatest pressure. As there are no channels from this point the oil cannot escape, and will support almost any load. There is no reason why one bearing only should be lubricated in all large bearings; it is desirable that each half should have its own supply of oil (Fig. 3).

Fig. 3.

In class C are included the bearings of horizontal engines, especially the main bearing next the connecting rod. This is the most troublesome kind of bearing, as it has double duty to perform. To facilitate the taking up of the wear, the device shown in Figs. 4 has been used. The bearing is in four parts, three of which can be drawn towards the center. The oil is introduced at the top at a point of least pressure, but before it arrives at the first point of greatest pressure it has to jump a joint. At some parts of the stroke it cannot do this, as the load is too great, but at others it can, and it is then carried into bearing No. 2. Before it can reach No. 3 it has another jump greater than the last, because when the oil escapes here it cannot return. No. 3 receives very little oil, but No. 4 gets even less, as the pressure is never off No. 3, and there is still another joint to jump. The only way of

meeting the difficulty is to have two complete bearings. One of these would be jointed at the top and bottom, and would take the thrust of the piston, the other would be jointed in the opposite

Figs. 4



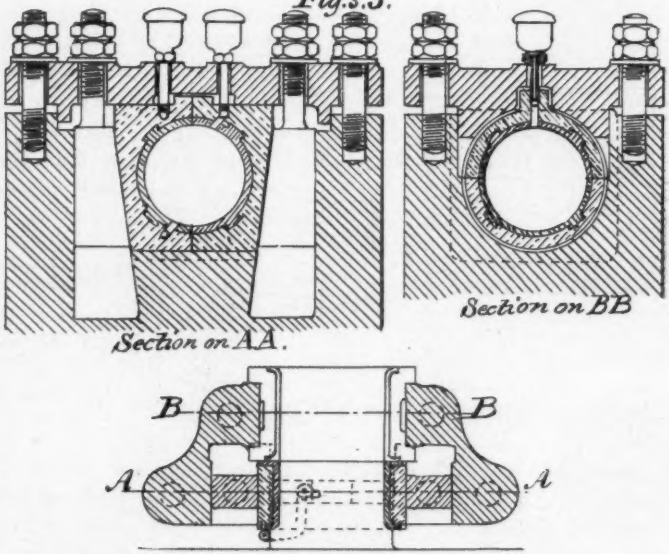
direction, and would take the weight of the shaft and fly wheel. Such a bearing is shown in Figs. 5. It is very important that the oil should be fed at a point exactly central in the length of the bearing. If it is fed a little to one side, the oil separates the surfaces and tilts the bearing, the consequence being that the oil is forced out at one end too fast, and at the other end the bearing is probably running dry.

The sides of a bearing should be carefully eased off, and should slope toward the center to bring the oil that escapes from the other half towards the center. Channels in the bearing or shaft surface can do no good except as receptacles for the debris when the bearing seizes. Such an advantage is very doubtful, but if they are used for this purpose they should be in line with the shaft, and not pass round it. They must not extend more than three-quarters the length of the bearing.

When it is fully recognized that a bearing will sustain a load exceeding one ton per square inch, and that most costly and serious troubles are experienced with bearings that are loaded only to a twentieth of that load, it must be admitted that this

subject requires investigation. The Author hopes that this paper may in some small degree have the effect of suggesting new

Figs. 5.



channels of thought and experiment that may add to knowledge on a subject of such great importance.

Appendix.

TABLE I.—RESULTS OF EXPERIMENT LVI.

Time.	Total load on top bearing.	Temperature.	Total tangential pull.	Time.	Total load on top bearing.	Temperature.	Total tangential pull.
Minutes.	*Pounds.*	° *Fah.*	*Pounds.*	*Minutes.*	*Pounds.*	° *Fah.*	*Pounds.*
...	5,376	98	...	80	32,906	230	470.0
10	5,376	113	80.0	90	37,632	242	400.0
20	7,706	116	115.0	100	42,470	245	350.0
30	12,723	128	155.0	115	42,470	242	400.0
40	16,934	134	157.5	125	42,470	262	400.0
50	19,264	154	215.0	135	42,470	264	400.0
60	23,834	220	400.0	145	42,470	258	350.0
70	27,552	182	350.0	150	47,846	280	470.0

Area of bearing surface of top bearing = 2½ inches by 8 inches = 20 square inches; bearings lubricated with sperm oil. Diameter of shaft, 4 inches; speed, 266 revolutions per minute.

After having run for 150 minutes, bearing was taken out; its bearing area was highly polished at the ends, and seemed to have seized at the middle.

EXPERIMENT LVIII.

The same bearings were used as in Experiment LVI, and they were lubricated with sperm oil and pads. The total load on the bearing was 15,132 lbs. = 756.6 lbs. per square inch. A vacuum gauge connected to the center of the bearing showed a vacuum of 28.4 inches of mercury, the barometer standing at 30.2 inches.

TABLE II.

Time.	Temperature.		Load.	Tangential pull.	Time.	Temperature.		Load.	Tangential pull.
	L. H.	R. H.				L. H.	R. H.		
	° *F.*	° *F.*	*Tons.*	*Lbs.*		° *F.*	° *F.*	*Tons.*	*Lbs.*
7.30	127	127	18.96	115.5	12.0	145	139	18.96	95.0
8.0	136	132	18.96	115.0	12.30	146	139	18.96	95.0
8.30	138	134	18.96	115.0	1.0	146	140	18.96	95.0
9.10	96	96	4.6	60.0	2.10	98	98	4.6	95.0
9.30	128	125	18 96	110.0	2.30	130	130	18.96	105.0
10.0	135	131	18.96	102.5	3.0	134	134	18.96	100.0
10.30	138	133	18.96	100.0	3.30	137	137	18.96	95.0
11.0	140	137	18.96	100.0	4.0	138	138	18.96	95.0
11.30	143	138	18.96	95.0	4.30	140	140	18.96	92.5

Area of bearing surface of top bearing = 18 square inches; bearings lubricated by pads hung on either side of the shaft and feeding neatsfoot oil by capillary attraction. Diameter of shaft, 4 inches; speed, 266 revolutions per minute.

THE BEARINGS OF THE MARINE ENGINE.

By John Dewrance, Member.

[Read before the Institute of Marine Engineers.]

The first object of this paper is to lay before the members an explanation of the way that oil enables a bearing to support its load, and afterwards to deal with the modifications in the design of the bearings of the marine engine that follow from a right understanding of the principle of lubrication.

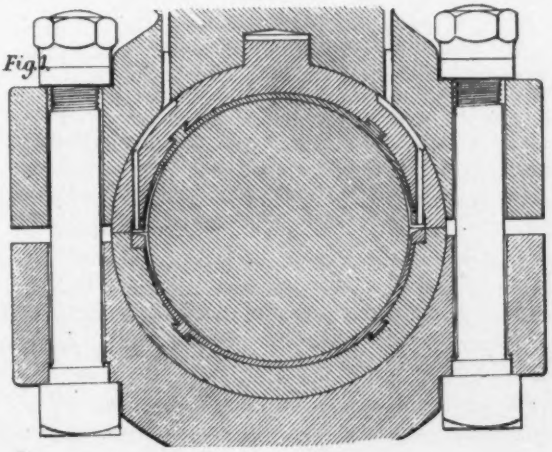
To simplify the explanation, the author has constructed a hand machine. The wheel, 24 inches diameter, represents a shaft, and the bearing may be said to have been flattened out into a plate. If the plate is laid upon the wheel and the micrometer adjusted, the hand will be observed to stand at a given mark. If oil is now put into the trough, so that it touches the wheel, it will be observed that the hand of the micrometer has moved, showing that the plate has risen. This means that between the surface of the wheel and the bearing there is a layer of oil on which the bearing is floating. If now we load the bearing, we must either squeeze out this layer of oil or the oil must be subject to a pressure sufficient to support the extra load. By means of a hole through the bearing and a pressure gauge, this point can very readily be proved. As will be seen, as the hole is brought to the point of contact between the bearing and the shaft the pressure on the gauge rises to a point that corresponds to the load put upon the bearing. As explained in a paper by the author, printed in the Proceedings of the Institution of Civil Engineers, the oil adheres to the surface of the wheel or shaft, and the force of this adhesion is multiplied by the incline formed by the bearing to the shaft. If the hole is put beyond the center of the shaft, the air in the hole adheres to the shaft and is carried round, leaving a vacuum, as shown by the gauge. Under favorable circumstances this vacuum has amounted to 30 inches, being within $\frac{1}{4}$ inch of the barometer at the time. The conclusion to be drawn from this

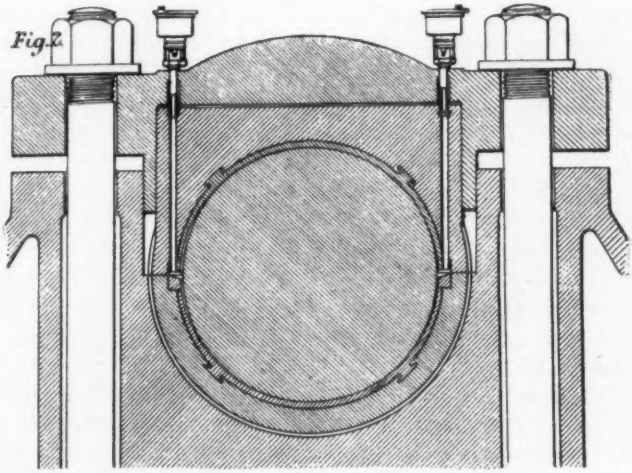
experiment is that bearing surfaces that are properly lubricated are separated by a film of oil at a pressure per square inch equal to the load that is upon them.

The realization of this fact suggests the following rule, which can be easily committed to memory, and is applicable to nearly all classes of bearing. Introduce the oil at the points of least pressure, and do not provide a means of escape for it at the points of greatest pressure. It is very easy to find out these points of an ordinary bearing that are least subject to pressure, and the oil can generally be brought there with a little scheming. The means of escape most generally met with are oil holes and channels that frequently occur just at the crown of the bearing where the pressure is greatest.

When the pressure on such a bearing is intermittent, the oil goes in when the pressure is taken off and escapes out again when the pressure comes on, the effect being that the bearing is only able to support a proportion of the load that it could support if lubricated according to the rule. It is quite impossible to lubricate such a bearing at all, if subjected to a continuous load. Not a drop of oil will run down the hole at the crown of the bearing, and if oil is put on the shaft elsewhere it runs out at the hole at the crown.

Fig. 1.





The diagrams represent the principal bearings of the marine engine to which this rule has been applied. Fig. 1 is the big end of the connecting rod of a vertical inverted marine engine. In this case the two sides are the points of least pressure, and, as will be seen, the oil is led to chambers at the sides. From these chambers suitable inclined planes are provided, and the oil will arrive at the surfaces which have to bear the load at a pressure per square inch equal to the load. It will be noticed that in each case the means of lubrication are in duplicate. This, in the case of large bearings, is strongly recommended, as otherwise one-half of the bearing is only lubricated by the oil that has passed through the other half, and in very large bearings this is not always found to be sufficient, especially when the bearings are new. The double supply of oil is also a great safeguard against failure.

Fig. 2 represents the crankshaft bearings. The duty of these bearings is almost identical with that of the big end of the connecting rod. As will be seen, the lubrication is also the same. It seems to be very generally the custom to make these bearings hexagonal or square on the outside. Such a bearing is very difficult to get out to examine or scrape up. The bearing shown is circular, and is prevented from turning with the shaft by a

square part on the top half. Such a bearing can be taken out very readily.

The several views in Fig. 3 represent the little end of the connecting rod. Here again the oil requires to be delivered to the sides the same as to the big end. The diagram shows the oil

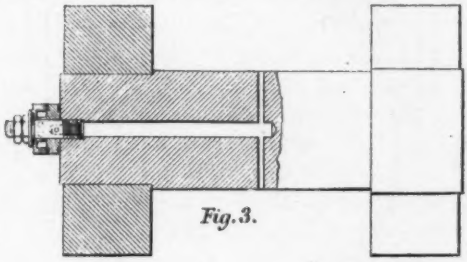
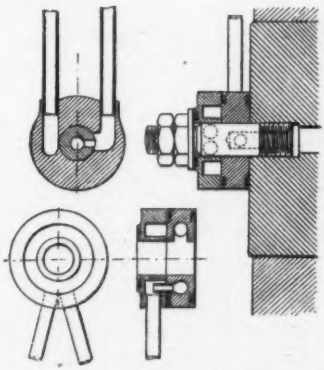


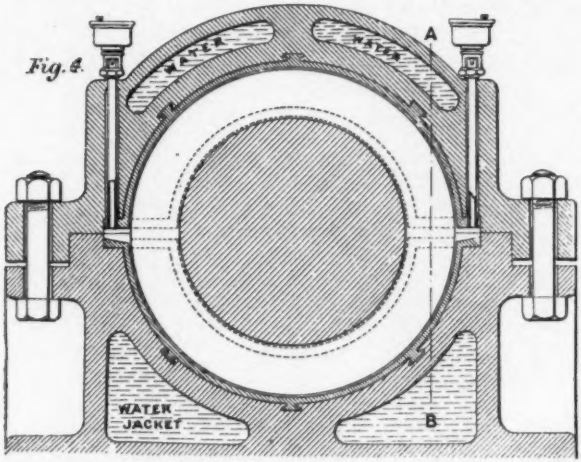
Fig. 3.



conducted through the pin. At the end there is a swivel joint. One of the pipes shown brings the oil for the little end and delivers it through the pin; the other pipe brings the oil for the big end and delivers it on the other side into a chamber from which it is conducted by two pipes to both sides of the big end.

Figs. 4 and 5 show the thrust block. It is strongly recommended that the cooling water should be kept away from the oil, as the mixture of water and oil is an inferior lubricant to pure oil. As will be seen, chambers are formed in the casting through

which water can be circulated. There is a difficulty about this form of bearing, in that there is no point of least pressure at



which to introduce the oil, so it is necessary to make one. To accomplish this the edges are sloped off, as shown on diagram

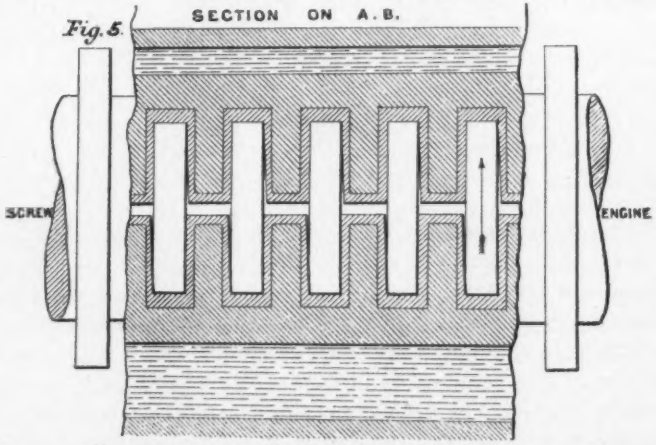


Fig. 5. With inclined planes such as are shown, it is possible to draw the oil in between the surfaces up to any pressure.

If once we accept the principle set out in this paper, it follows of necessity that a hot bearing is due to a failure of lubrication. If the oil is supplied, as shown by the diagrams, this may be due to the fact that the shaft is not round, is not running true, or else, that it is not smooth enough. Shafts should be finished by clamps lined with emery cloth till they are well polished. Another cause may be that the bearings are not properly fitted. It is no light task to surface up a bearing so that the shaft beds thoroughly.

Having dealt with the subject of the means of lubrication, the next point is the oil. In these days of competition and lowest tender, very inferior oils are sometimes used in very superior ships. Adulteration of oils is so general that the names by which they used to be known have no longer any real meaning. Whenever it is possible, it is the safest way to have the samples of oil examined and reported upon by a chemist, who also has the means of testing the lubricating properties.

Another very important point is the material of which the bearing is composed. Marine engineers seem to be very generally agreed that the bearing should be lined with one of the alloys known as the white metals. These may be divided into three classes. The first contains anything up to 80 per cent. of tin, the second anything up to 80 per cent. of zinc, and the third anything up to 80 per cent. of lead. If we could make sure of always using perfectly neutral oil there would be very little to choose between these three classes, but in order to obtain a neutral oil, such as is prepared for clockmakers, the oil is agitated with zinc and lead shavings, a portion of each is converted into zinc or lead soap, which is afterwards separated from the oil. The principal impurity of lubricating oils is oleic acid, which rapidly corrodes lead and zinc. The effect then of using an alloy that contains a large proportion of lead or zinc is that the impurities of the oil combine with the surface of the bearing. Of course this may be avoided by using a very pure oil, but those that are responsible for the lining of the bearing are not always responsible for the quality of the oil used, and even if they are at the time, they cannot make sure that they will always have it under their control. Tin is not affected by oleic acid or any of the impurities of oil, so the safest

way is to use an alloy composed principally of tin, and only containing enough of the most suitable metals to harden it to the utmost. Many of the alloys at present used are too soft and yield with a load of even as little as a quarter of a ton to the inch. Such a metal is liable to squeeze out in use. An all-round lining metal ought to stand at least 5 tons to the square inch without any yield. The best alloys will stand 8 tons.

It is a very general custom among marine engineers to hammer the alloy after the bearing is lined. If the alloy is as hard as is desirable it cannot be very ductile, and this hammering cracks it in all directions; if, on the other hand, the alloy is ductile enough to stand hammering, it is conclusive proof that it has too low a compression test to be suitable for lining marine bearings. The author feels considerable diffidence in appearing to lay down the law to those who have had so much more experience of marine bearings. It is, however, right to say that the points dealt with are no untried ideas, but are the result of many years of study of the subject and a very long series of experiments. It would take too long to even outline these experiments, but they entirely confirm the facts laid before you in this paper.

NOTES.

RUN OF THE U. S. S. *CASTINE*.

This steel twin screw gunboat of 1,210 tons displacement (at a mean draught of 12.25 feet) and 1,600 I.H.P. arrived at Norfolk, Va., on November 7, 1896, from the South Atlantic Station. This gunboat was described on page 846 of volume V and page 176 of Volume VI of the JOURNAL.

The homeward run was divided into three parts: from Montevideo, Uruguay, to Bahia, Brazil; thence to the Island of St. Lucia; and thence to Norfolk. The distances as run and the times were 1,782.3 nautical miles in 7 days 19 hours, 2,438.5 miles in 10 days 12 hours, and 1,396 miles in 5 days 18 hours, respectively.

For data of this run the JOURNAL is indebted to the kindness of Chief Engineer W. F. Worthington, U. S. N.

The ship's bottom, when leaving Montevideo on October 10, was perfectly clean. Two boilers were in use, except for about six hours, during which a leaky tube in one boiler was plugged. No deductions are made for this, nor for the times that only one engine was in use, one hour while adjusting a main bearing, and two hours while adjusting an eccentric strap. Both engines were stopped one and one-half hours while a leaky condenser tube was plugged.

The evaporator and dynamo were in operation every night in addition to all other auxiliaries, except the fire room blowers.

The coal for the first two parts of the run was Ocean Merthyr, of good quality for the first, and fair for the second part. For the last part, Pocahontas coal of fair quality was used. The quantity of coal burned during each hour of the entire run was constant, 1,340 pounds. The average mean draught for each part of the run was 12 feet, the corresponding displacement being 1,177 tons.

First part, Montevideo to Bahia, 1,782.3 miles. Total steaming time, 7.792 days; average speed, 9.53 knots; average number of revolutions per minute, 128.9. Miles run per ton of coal, 16; pounds of coal per mile, 140. I.H.P., main engines, 420; auxiliaries, 40. Time under sail, 16 per cent.

Second part, Bahia to St. Lucia, 2,438.5 miles. Total steaming time, 10.5 days; average speed, 9.67 knots; average number of revolutions per minute, 128.2. Miles run per ton of coal, 16.2; pounds of coal per mile, 138. I.H.P., as above. Time under sail, 45 per cent.

Third part, St. Lucia to Norfolk, 1,396 miles. Total steaming time, to noon of the day before arrival, 5.75 days; average speed, 10.1 knots; average number of revolutions per minute, 134.4. Miles run per ton of coal, 16.9; pounds of coal per mile, 132. I.H.P., main engines, 456; auxiliaries, 40. Time under sail, 76 per cent.

Comparison may be made between the run of the *Castine* and that of the *Bancroft* and the *Petrel*, the latter described on page 581 of Vol. VIII. It should be added that the *Petrel* was under sail 75 per cent. of the time between Yokohama and Honolulu, and 17 per cent. from Honolulu to San Francisco.

THE TRANS-ATLANTIC RUN OF THE U. S. S. *BANCROFT*.

For the following information of the run of the Naval Academy Practice Vessel *Bancroft*, the JOURNAL is indebted to the kindness of P. A. Engineer E. T. Warburton, U. S. N. This vessel was fully described on page 365, Volume V.

The *Bancroft* was docked and refitted in September, at the Navy Yard, New York, in ten days, for a cruise on the European station.

The three masts were replaced by two, and the barkentine rig changed to schooner rig. Changes were made in the storeroom to provide additional bunker space, and much other work was done in the short time, by working day and night.

The *Bancroft* left Tompkinsville, Staten Island, N. Y., September 15, 1896, using both boilers, and arrived at Fayal, Azores, September 25. The dynamo was run about 12 hours, and about

300 gallons of water distilled every day. The vessel started with about 160 tons of bituminous coal on board. Distance steamed, 2,133 nautical miles; time, 10 days 1 hour, or 10.04 days; average speed, 8.85 knots; average revolutions per minute, 124. Coal used for all purposes, 102.69, coal per day, 10.22 tons; miles per ton of coal, 20.77. There remained in the bunkers 57.5 tons.

Left Fayal September 28, and arrived at Gibraltar, October 4. Strong head winds were encountered for the greater part of the distance. Distance steamed, 1,136 nautical miles; time, 6 days 3.75 hours, or 6.16 days; average speed, 7.69 knots; average revolutions per minute, 129.2. Coal used for all purposes, 59.33 tons; coal per day, 9.63 tons; miles per ton of coal, 19.14.

Left Gibraltar, October 6, and arrived at Smyrna, Asia Minor, October 15. Distance steamed, 1,631 nautical miles.

The coal obtained at Gibraltar was of such very inferior quality that the daily consumption was necessarily increased.

The trip from Tompkinsville to Smyrna (4,900 nautical miles), in 29.5 days, including stoppages of 2½ days at Fayal, and 2 days at Gibraltar, at an average speed of 8.25 knots, and an average daily coal consumption of 10.3 tons, is not a bad showing for this little vessel. She was not intended for such long cruises, and in previous service had only made two or three-day passages along the coast.

TESTS FOR STEEL BOILER TUBES.

The following specifications for steel boiler tubes, as used by Messrs. John I. Thornycroft & Co., Chiswick, for much of their work, were kindly furnished to the JOURNAL by Mr. John Platt, New York, the Agent in the United States for the Messrs. Thornycroft. Their reasons for using steel is that a product of the quality necessary to fulfill the specifications has been made specially for this work, and that no one in England has made any cold drawn charcoal iron tubes. For their continental work, they have used cold drawn charcoal iron tubes very extensively, as certain countries specify that these shall be used. The Messrs.

Thornycroft always insist that the tests, according to the specifications, must be rigidly enforced, as they have found that, in order to minimize local corrosion and insure a long life to the tubes, it is necessary to keep fully up to this standard in all cases. The tests required by the British Admiralty were given on page 814 of Volume VIII.

Tests for Boiler Tubes.—The boiler tubes are to be solid drawn, finished cold, so as to remove all traces of the hot process, and leave perfectly smooth surfaces inside and outside.

The tubes are to be perfectly straight, smooth, cylindrical, of uniform sectional thickness, and of equal diameter throughout, excepting as specified below; they are to be free from any scale, longitudinal seaming, grooving or blistering, either internally or externally, and they are not to be oiled, varnished or painted.

The ends of the tubes are to be pressed or squeezed up for screwing in a die or press (not by hammering or upsetting.) Any case discovered of a strip being welded on will render the whole liable to rejection. The tubes are to be made from acid or basic open hearth steel, which material is to stand the following tests:

	Ultimate tensile strength.		Elongation in a length of 2 inches, per cent.
	Not more than, tons *	Not less than, tons.*	
(a) Annealed pieces cut from the forging from which the tubes are to be made	24	21	33
(b) Annealed pieces cut from the tubes....	26	...	27

*Tons of 2,240 pounds.

Test (a) is to be carried out at the works of the tube maker, and (b) at the works of the makers of the boilers.

Strips cut from the tubes, flattened, heated to a blood heat, and plunged into water 82 degrees Fahrenheit temperature, should be capable of being doubled over a radius of $\frac{1}{2}$ inch without fracture. This test is to be carried out at the works of the

makers of the boilers. Pieces two inches long, cut from the tubes under $\frac{3}{16}$ inch thick are to be capable, when cold, of being hammered down lengthwise until their length is reduced to one inch. The tubes themselves are to be capable of being flattened by hammering at any part until the sides are close together, for tubes under $\frac{3}{16}$ inch thick, and for tubes $\frac{3}{16}$ inch thick and over the sides are to be brought to a distance apart of twice the thickness of the material of tubes; in each case without fracture. The ends of the tubes under $\frac{3}{16}$ inch thick are to admit of being expanded cold by a three-roller expander, worked in a series of three tube holes, and hot by a solid drift, to the following increases of diameter:

With roller expander fitted with three rollers (cold), 12.5 per cent.; with solid drift (hot), 20 per cent.

Tubes $\frac{3}{16}$ inch in thickness and over are to admit of being expanded hot and cold to half the increase in diameter required for the tubes under $\frac{3}{16}$ inch.

The above tests are to be carried out at the works of the makers of the boilers, and are to be applied to two per cent. of the tubes, to be selected by the examining officers, after electric galvanizing; and these tubes are to be completely destroyed for the purpose of this test. The tubes for this purpose are to be arranged in parcels of one hundred, and all rejected tubes are to be marked by the examining officers, so that they may be capable of identification. The failure of the tubes selected will reject the parcel of one hundred from which they were taken.

Each tube must stand a test by water pressure internally of 1,500 pounds per square inch, after bending and annealing, but before electro-galvanizing, without the slightest indication of weeping. This test to be carried out at the boiler makers' works.

The failure of a large proportion of the tubes selected to stand any of the above tests in a satisfactory manner will render the whole of any delivery liable to rejection.

All tubes are to be thoroughly pickled to remove scale, and unless otherwise ordered are to be galvanized externally by electro-deposition. Each tube is to be removed from the electro-

depositing bath after being immersed and subjected to the electro-depositing action for about fifteen to twenty minutes. It is then to be carefully examined, and if any defects are exhibited the tube is to be rejected. The satisfactory tubes are to be replaced in the bath and coated with zinc to an extent of 1¼ ounces per square foot of galvanized surface.

The limit of variation of weight of each tube which will be allowed is 2½ per cent. above the weight calculated from the approved dimensions, allowing 480 pounds per cubic foot of metal. All tubes exceeding this limit or less than the specified thickness will be liable to rejection. All tubes which are less than the external diameter specified, or more than 1 per cent. larger than the diameter approved will be rejected.

FRANCE.—SHIPBUILDING PROGRAM, 1897.

The following vessels, the construction of which has been approved by the Budget Commission, are to be laid down during 1897.

One triple screw battleship of about 8,860 tons displacement, to be built at Brest. Speed, 17 knots.

One first class, triple screw, sheathed cruiser of about 11,050 tons displacement, to be built at Toulon. Speed, 23 knots.

Two first class, triple screw, sheathed, station cruisers of about 5,410 tons, one to be built at Lorient and the other by contract. Speed, 23 knots.

One third class station cruiser of about 2,410 tons displacement, to be built by contract. Speed, 20.5 knots.

One sheathed gunboat of about 620 tons displacement, to be built by contract. Speed, 13 knots.

One torpedo boat destroyer of a new type. Speed, 26 knots.

Two first class torpedo boats of the Normand type and of about 84 tons displacement. Speed, 23.5 knots.

All guns under 7.64 inches caliber will be of the rapid fire type.

Particulars of the above new vessels, compiled from the "Yacht," Paris, and other sources are given under "Ships."

GERMANY.—SHIPBUILDING PROGRAM FOR 1897.

According to "The Times," London, the following vessels are to be laid down.

One battleship to replace the *König Wilhelm*. This is to be of the same type as the *Kaiser Friedrich III*, described on page 827 of Vol. VII, and in the table facing page 556 of Vol. VIII.

Two second class cruisers; a despatch vessel to replace the *Falke;* two gunboats to replace the *Hyäne* and *Iltis*, the latter having been lost off the Shantung Peninsula, China, on July 23, 1896; one torpedo vessel and eight torpedo boats.

HOLLAND.—SHIPBUILDING PROGRAM.

[From "Engineering," London, and "Mittheilungen aus dem Gebiete des Seewesens," Pola.]

According to the published program for the reorganization of the Navy, extending over a period of thirteen years, twelve protected cruisers will be built of the same type and speed, 23 knots, as the *Holland*, *Friesland* and *Zeeland* now building, (see JOURNAL, page 829, Volume VII, and pages 406 and 792, Volume VIII), except that the armor shields for the 5.9-inch and 4.7-inch guns will be 5.9 inches, and those for the 2.95-inch guns, 2.95 inches thick. The bow torpedo tubes will be under water, and bilge keels will be fitted. Other improvements will bring the cost of the new vessels up to $1,205,100 each.

Six armored vessels are also proposed of the same type as the *Piet Hein*, described on page 194 of Volume VIII. Some modifications will be made in the armament. Instead of three 8.27-inch guns of 32 calibers, there will be two 9.45-inch guns of 40 calibers, each in a barbette tower; the two 5.9-inch rapid fire guns will be replaced by four 4.7-inch rapid fire guns, protected by closed shields, 1.97 inches thick. Besides these, there will be several 2.95-inch rapid fire guns protected by shields. The displacement will be 3,874 tons on a mean draught of 17.5 feet. The engines, of 5,227 I.H.P., are to give the vessel a speed of at least 16 knots.

For coast defense, three monitors, type A (larger type, about the same type as the *Reinier Claessen*), and three monitors, type B (smaller type) are proposed, together with fifteen gunboats, fifteen torpedo boats, type A (30 knots), six torpedo boats, type B (23 knots), and ten torpedo boats, type C (18 to 20 knots.) For protection of the fishermen three schooners are proposed.

The monitors, type A, will have a displacement of 1,476 tons on a draught of 10.17 feet; a protective deck 1.97 inches thick; two 8.27-inch, 40 caliber, guns in two barbette towers, protected by 7.87-inch steel armor, the guns having 5.9-inch shields; four 2.95-inch rapid fire guns, protected by .98-inch shields, and four 1.46-inch rapid fire guns. The engines are to develop 690 I.H.P., and to give the vessel a speed of at least 9.5 knots. The bunker capacity will be about 60 tons.

The monitors, type B, will have a displacement of 1,384 tons on a draught of 9.51 feet; a protective deck, 1.97 inches thick; one 8.27-inch, 40 caliber, gun in a barbette forward, protected by 7.87 inches of armor, the gun having a 5.9-inch shield; one 4.7-inch, 40 caliber, rapid fire gun, aft, with a closed shield 1.97 inches thick; four 2.95-inch and four 1.46-inch rapid fire guns, the former protected by .98-inch shields. The I.H.P. will be 670; the speed, 9.5 knots, to be maintained for four hours; and bunker capacity, about 60 tons.

The gunboats will have a displacement of 467.5 tons on a draught which is not to exceed 8 feet 4 inches; a protective deck .98 inch thick; four 2.95-inch and four 1.46-inch rapid fire guns, the former protected by .98-inch shields. The I.H.P. will be about 550; the speed, 11.75 knots, to be maintained for four hours; and bunker capacity, 23 tons.

The torpedo boats, type A, will be of the same general design as the French *Forban*. They will have 128 tons displacement; two 1.46-inch rapid fire guns; two torpedo tubes; a speed of 30 knots; and a bunker capacity of 15 tons. Torpedo boats, types B and C, will be the same as the Dutch boats A and Nos. *XXI* and *XXII*, respectively.

The estimate for the building of these new vessels is 55,445,000 guilders, or $22,843,340, which is to be spread over thirteen

years, the sum of $2,060,000 being expended for each of the first eleven years, and the balance in the last two years.

Estimated cost of each cruiser,	$1,205,100
armored vessel,	1,499,680
Monitor, Type A,	626,620
Monitor, Type B,	527,740
gunboat,	144,200
torpedo boat, Type A,	189,520
torpedo boat, Type B,	70,040
torpedo boat, Type C,	24,720
schooner,	195,700

JAPAN.—SHIPBUILDING PROGRAM.

The ships to be laid down during the current year are as follows.

Four battleships. Of these, one will be built in Japan and the other three in England. The latter are to have Belleville boilers and will follow generally the design of the *Magnificent*.

Four first class cruisers, two of which will be built in Japan. The other two are already in hand in private yards of the United States, and are described under "Ships" in this number.

Three second and two third class cruisers, of which only one second class cruiser will be built abroad.

Three torpedo gunboats, one torpedo tender and eleven torpedo boat destroyers. Of these, two gunboats and seven destroyers will be built in Japan.

Twenty-three first class torpedo boats, of which four will be built abroad.

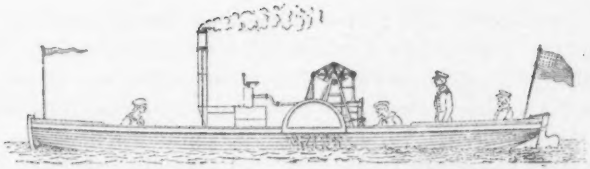
THE FIRST U. S. NAVAL STEAM LAUNCH.

In a new book, entitled "Reminiscences of an Octogenarian of the City of New York (1816–1860)," by Chas. H. Haswell, Esq., an Honorary Member of this Society, the following interesting accounts are found. The JOURNAL is indebted to the author for the cut and the subject matter.

"The first operating locomotive introduced into this country was one which had been procured in England by Horatio Allen, and it was put in operation at the West Point Foundry Shop in Beach Street, in April, 1829. Its power was estimated at nine horses; pressure of steam, 60 pounds per square inch; and its capacity, 5 miles per hour with a train of from 60 to 80 tons."

"June 4, 1829, the magazine of the steam frigate *Fulton* (the *Flogobombos* of Dr. Samuel L. Mitchell), in service at the Navy Yard, Brooklyn, as receiving-ship, exploded and killed one lieutenant and twenty-three marines, wounding seventy-two of the crew, with six missing. I was at the time on board of the steamboat *Citizen*, then in process of construction at the head of Water Street, witnessed the explosion, and visited the wreck immediately afterward. The *Citizen* was the second steamboat Captain Vanderbilt owned (the *Bellona*, built in 1816, a gift from William Gibbons, being the first), and the first that he built. The first he commanded was the *Stoughtenger* (in derision she was called 'The Mouse out of the Moutain'), 75 feet in length, propelled by what were termed paddles, but they were strictly *palmipedes*, in order to evade the Fulton claim for side wheels. This attempt was a signal failure. The *Bellona* was fitted with like propulsion, but it being condemned as useless, she was fitted with side wheels, and from this arose the litigation between William Gibbons and Gouverneur Ogden, in which Daniel Webster was engaged, as to the exclusive right of Fulton and his associate Livingston to steam navigation on the waters of the Hudson, a claim which the U. S. Supreme Court declared to be invalid."

"The first steam launch was designed in 1836 by and constructed in 1837 under the direction of the writer in this year, at the New York Navy Yard, and named the *Sweetheart*. On her trial trip and several succeeding, she was hailed and saluted by the bells of passing steamboats and by cheers from people who rushed to the ends of the piers to witness the novel sight. She attained a speed at the rate of 5.5 miles per hour. The engine was subsequently transferred to the first U. S. Naval School, then in Philadelphia."



From "Reminiscences of an Octogenarian." Copyright, 1895, by Harper & Brothers.

THE FIRST STEAM LAUNCH DESIGNED AND CONSTRUCTED AT U. S. NAVY YARD, NEW YORK.

Length 35 feet; beam, 4.25 feet; and depth, 3 feet. Engine, 4x12 inches. Wheels, 3.5 feet. Boiler, horizontal fire tubular. Scale, 120th part.

"April 25, 1845, the steamboat *Empire*, of the New York and Albany Line, on her passage to this city in a dense fog, ran into the pier, solid ballasted crib work, at the foot of Nineteenth Street, for the full length of 20 feet. A report of the occurrence was held to be so wholly at variance with the generally entertained opinion as to the practicability of such a result that many persons proceeded to the pier and measured the distance. The effect of such an impact upon like work is to this day a marvel."

TORPEDO BOAT *TURBINIA*.

This boat was built at Wallsend-on-Tyne by the Marine Steam Turbine Co., Newcastle, for the purpose of testing the application to marine propulsion of Mr. C. A. Parsons' steam turbine. The boat is of steel, and is 100 feet long over all, has a beam of 9 feet, and displaces about 42 tons when fully loaded. There is one water tube boiler with a grate surface of 42, and a heating surface of 1,100 square feet. The working pressure in the boiler is 225 lbs. per square inch, which is reduced to 150 pounds at the turbines. The closed fire room system of forced draft is used. The weight of the motor is given as 4.5 tons. On December 15, 1896, the boat made several runs over the measured mile outside the Tyne, a speed of 29.6 knots being attained with about 2,400 revolutions per minute of the motor.

EVAPORATIVE TEST OF AN ALMY WATER TUBE BOILER.

The following abstract is from the report of the tests made by Mr. Geo. H. Barrus for the Almy Water Tube Boiler Co., Providence, R. I., from October 30 to November 2, 1896.

The boiler, which has been in use about fifteen months in the shop, is of Class D size. The dimensions of the outside casing are: length, 51⅝, breadth, 51⅝, and height to top of casing, 78 inches. The grate is 40.1 inches long and 40.9 inches wide, and the surface, 11.38 square feet. The grate bars are $\frac{9}{16}$ of an inch wide and separated by ½ inch air spaces. The number of 1-inch pipes (two pipes to a section) which make up the length of the boiler, is twenty-six, and the total exterior heating surface, without allowance for portions more or less covered and unexposed to direct heat, is approximately 474 square feet. The area of opening for draft between the tubes is 4.6 square feet. On November 2, previous to the test, it was reduced by fire brick to one-half this area, or one-fifth of the grate surface. The drum is 12¾ inches in diameter and 48 inches long. The estimated weight of iron in the boiler, not including the covering, is 3,097 pounds, and of the contained water at ordinary level, 650 pounds. The stack is 19 inches in diameter, its height above the grate, 35.5 feet, and is fitted with a damper. The covering consists of wrought iron plates lined with layers of asbestos and fire brick. The feed water is heated by a coil of pipe, termed the "heater," in the upper part of the boiler chamber next to the stack. All steam not required for the shop work was allowed to escape through the safety valve to the atmosphere. During the tests there was a continual discharge of steam at this point.

The tests were made with damper wide open and with the maximum draft available. Those made November 2 contained two short runs with a steam jet in the stack. After completing the natural draft trial on that day, the fire was rekindled and then the two short runs were made.

The tests, excepting the two short ones, conformed to the standard method of the American Society of Mechanical Engineers; that is, the boiler was first heated to normal working condition, this operation consuming about one and one-half hours. The fire was then drawn and the new fire kindled. At the end of the run, the fire was burned down and finally hauled. On the two runs with forced draft, the tests were commenced and finished

under running conditions, the thickness of the fire at beginning and end being estimated. These trials are too short for absolute reliability as to the coal measurements. They were intended mainly to show the capacity of the boiler in horse power under the conditions of forced draft.

The feed water was drawn from two measuring barrels, one being placed on scales above the other, where it was weighed, the feed pump taking it from the lower barrel. Half-hourly observations of steam pressure, temperatures, weights, &c., were taken.

The quality of the steam escaping through the safety valve pipe was tested with the writer's calorimeter, 1895 pattern. The coal used on all tests was George's Creek Cumberland, and was commercially dry, the total heat per pound of coal being 14,168 B.T.U. The coal was sampled on each run, and dried for determining moisture.

On a preliminary run on October 30, a greater thickness of fire was maintained than on the later tests, and the results of the gas analysis showed the presence of a comparatively larger amount of carbonic oxide, which sometimes reached 16.9 per cent. of the total volume of gas. It is interesting to note this fact because the thickness of the fire was not excessive, seldom exceeding more than eight inches. The evaporative result conforms to that anticipated from the gas analysis, being only three-quarters of that obtained on the main tests.

Conclusions.—It appears from these results, that, in point of economy, the boiler compares favorably with the best types. An evaporation of 11.922 pounds of water per pound of combustible, obtained on November 2, is rarely exceeded by any form of hand-fired water tube boiler, whatever its size. The dryness of the steam was a noticeable feature. Even with forced blast, when over 7 pounds of water were evaporated per square foot of surface per hour, the moisture was less than 1 per cent.

Practically all of the heat units available in the coal were accounted for, either in useful evaporation, or in chimney and other wastes.

Conditions as to capacity.....	Normal.	Normal.	Forced draft. 1-Heavy.	Forced draft. 2-Medium.
Date of test—1896.....	Oct. 31.	Nov. 2.	Nov. 2.	Nov. 2.
Duration of test, hours.....	9.15	6.1	2.03	2.0
Coal consumed per hour.....lbs...	139.2	159.2	409.4	271.5
Coal consumed per hour, per square foot of grate.....lbs...	12.2	13.99	35.98	23.86
Per cent. of ashes and clinkers.....	8.6	10.3	...	...
Water evaporated per hour.....lbs...	1,238.5	1,403.6	3,231.	2,207.
Equivalent evaporation per hour, feed 100 degrees, pressure 70 pounds .....lbs...	1,300.4	1,472.8	3,389.3	2,315.1
Horse power developed on basis of 30 pounds.....H.P...	43.3	49.1	112.98	77.2
Equivalent evaporation per square foot heating surface per hour,lbs...	2.74	3.1	7.15	4.88
Average boiler pressure.....lbs...	147.1	146.8	153.6	140.1
Average temperature of feed water, ...degrees...	56.7	56.6	56.	56.
Average temperature of flue-gases, ...degrees...	513.	473.	850.*	715.*
Average draft suction.....inch...	.12	.14	.71	.4
Per cent. of moisture in steam.....	0.35	0.4	0.72	0.42
Water evaporation per pound of coal.....lbs...	8.89	8.87	7.89	8.13
Equivalent evaporation per pound of coal from and at 212°.....lbs...	10.74	10.69	...	...
Equivalent evaporation per pound comb. from and at 212°.....lbs...	11.75	11.92	...	...
Efficiency of combustible..per cent...	...	75.8	...	...

* Only one observation. The actual temperature on the heavy forced draft test was higher than 850 degrees, the limit of the thermometer used—probably 900 degrees.

LIQUID FUEL ON RUSSIAN TORPEDO BOATS.

The tests made with liquid fuel on the torpedo boat *Viborg* during October, 1896, proved so satisfactory and showed such advantages that all torpedo boats having locomotive boilers are to be fitted to burn *masut* as well as coal. The torpedo boats having water tube boilers will not be so fitted until tests have been made on the six boats now building and which are to have Du Temple boilers fitted for liquid fuel. Part of the present coal bunkers will be arranged into compartments which will have a capacity of 3.25 tons of *masut*. The trials on the *Viborg* showed that it was very important to give special attention to the first filtration of the *masut*.

SHIPS.

UNITED STATES.

San Francisco.—On run from Genoa, Italy, to Smyrna, Turkey, trials were made with one and two screws, each lasting forty-eight hours.

The coal used in both trials was Albion Merthyr, Welsh, of good quality, and made very little refuse. The vessel was docked last in May 1, 1896.

The trial with one screw was made from Oct. 8 to 10, 1896, and with two screws, from Oct. 11 to 13. The wind was very light and the sea smooth during both trials. Two boilers were used on both trials, the grate surface having been reduced to about two-thirds of the whole by covering the backs of the bars with ashes. The grate surface actually in use was 184 square feet, and the heating surface, 9,828.8 square feet.

The excessive boiler power for the I.H.P. developed probably accounts for a part of the high values of the coal per I.H.P. per hour. From two previous trips, made with one boiler and two screws, it had been found that for a speed of about 7.4 knots, the number of pounds of coal per nautical mile was only 250, as compared with 305 and 310 pounds for the present trials. In the opinion of the chief engineer, it would be more economical, when running at low speeds, to use one boiler with assisted draft, if necessary.

The following table gives the results of the October trials.

	One screw.	*Two screws.*
Steam pressure in boilers, pounds per square inch	107	107
Vacuum, inches	26.6	27
Revolutions per minute, starboard screw	65.03	52
port screw	...	52
Speed, by observation, knots	8.29	8.51
Slip of propellers, per cent	31.09	11.54
Helm angle, while using one screw, degrees	8 to 9	...
I.H.P., main engines	820.58	695.14

	One screw.	*Two screws.*
I.H.P., auxiliaries	52.	71.
total, (A)	872.58	766.14
Coal per hour, main engines, dynamo, flushing pumps and steering engine, pounds	2,542.35	2,644.85
Coal per day, for above purposes, tons	26.91	27.98
distilling and galley, tons	1.62	1.62
total	28.53	29.60
Days' steaming on bunker capacity (630 tons)	22.08	21.28
Knots per ton of coal used for all purposes	6.97	6.90
Coal per I.H.P. (A) per hour, pounds	2.91	3.45

Newport and **Vicksburg.**—These two composite gunboats, described on pages 581 and 804 of Vol. VII, and page 799 of Vol. VIII, were successfully launched at the Bath Iron Works, Bath, Maine, on Saturday, December 5, 1896. The *Newport* was Gunboat No. 12 and the *Vicksburg*, Gunboat No. 11.

Torpedo Boats Nos. 4 and 5.—These torpedo boats, built by the Columbian Iron Works and Dry Dock Co., Baltimore, were launched on November 10, 1896, and January 6, respectively. The description of the sister boat was given under *Torpedo Boat No. 3* on page 799 of Vol. VIII.

Annapolis.—This composite, single screw gunboat, formerly No. 10, described on pages 581 and 804 of Vol. VII, and page 799 of Vol. VIII, was successfully launched at the Crescent Shipyard, Elizabethport, N. J., on Wednesday, December 23, 1896.

Manning.—The second of the revenue cutters, fully described on page 561 of Volume VII, was successfully launched by the builders, The Atlantic Works, East Boston, Mass., on Friday, January 22.

McCulloch.—This revenue cutter, some particulars of which were given at the foot of page 565 of Volume VII, was successfully launched by the builders, The Wm. Cramp & Sons' Ship and Engine Building Co., Philadelphia, Pa., on Saturday, December 19, 1896. The vessel is intended for use on the Pacific Coast.

ARGENTINE REPUBLIC.

Corrientes.—The last of the four destroyers, built by Messrs. Yarrow & Co., Poplar, completed satisfactory official trials on

Nov. 18, 1896. A complete description of this class of destroyers was given under the *Santa Fé* on page 609 of Vol. VIII. The following particulars of the trial are from "Engineering," London.

The draught aft was 4 feet 11 inches, forward, 3 feet 11 inches, mean, 4 feet 5 inches. The contract load of 35 tons was on board. The vessel started with 22 tons of coal, and burned during the three hours of the trial 10.65 tons. There were 56 persons on board.

The mean speed for the six runs over the mile was 27.359 knots, the revolutions per minute being 372. During the three hours, the mean steam pressure was 151 pounds per square inch; revolutions per minute, 372.7; speed, 27.41 knots.

The engines, boilers and the automatic feed regulator worked perfectly.

The coal consumption for the four destroyers varied from 2 to 2.25 pounds per I.H.P. per hour.

Entre Rios and **Santa Fé.**—On their passage to Buenos Aires, experiments were made to find the most economical system for long runs at low speed. It was found that a saving in fuel of about 10 per cent. was secured by running with one engine alone at a speed of from 10 to 11 knots, the other propeller being uncoupled. "The Engineer," London.

Entre Rios made the run from St. Vincent to Buenos Aires, a distance of about 3,700 nautical miles, without a stop.

AUSTRIA.

Wien.—This coast defense vessel, described on page 610 of Volume VIII, completed her trials in the latter part of 1896. With the engines developing 8,384 I.H.P., the speed was 17.6 knots. She was launched on July 6, 1895.

BRAZIL.

Marshal Deodoro and **Marshal Floriano.**—These twin screw coast defense vessels are building by the Forges et Chantiers de la Mediterranée at La Seyne, and were ordered in 1896. They will have a displacement of 3,160 tons and a speed of about 14 knots. The I.H.P. will be about 3,400.

Amazonas.—This steel, sheathed cruiser, built by Sir W. G. Armstrong & Co., at their Elswick yard, Newcastle, was successfully launched on Friday, December 4, 1896. She is a sister vessel of the *Barrozo* described on page 806 of Vol. VIII.

The length of the vessel is 330 feet; beam, 43.75 feet; mean draught, 16 feet 10 inches; displacement, 3,450 tons.

There is a curved steel protective deck extending the whole length of the vessel. It is 1¼ inches thick on the horizontal part and 3½ inches thick on the sloping parts.

The armament will consist of six 6-inch and four 4.7-inch rapid fire guns, all 50 calibers long and of the Elswick pattern; ten 6-pounder and four 1-pounder Nordenfeldts, four Maxims, two field guns and three torpedo tubes. Of the 6-inch guns, three are arranged to fire ahead and three astern.

The twin screw engines are designed for 7,500 I.H.P. and to give the vessel a speed of from 20 to 20.25 knots.

The bunker capacity is 800 tons, the side bunkers extending to 6 feet above the water line.

Tupy.—The second one of the three torpedo gunboats building by the Germania Co., Kiel, Germany, was launched on November 15, 1896. She is a sister vessel of the *Caramuru*, described on page 611 of Volume VIII. The data are here repeated.

The length is 259.1 feet; beam, 30.84 feet; draught of water, 10.17 feet; displacement, 1,014 tons.

The armament will consist of two 4.13-inch, six 2.24-inch and four 1.46-inch rapid fire guns; and three 17.72-inch torpedo tubes, one in the bow and two on deck.

There are two triple expansion engines, which, under forced draft, are guaranteed to develop 5,918 I.H.P., and to give the vessel a speed of 23 knots. There are five large locomotive boilers.

CHILI.

Esmeralda.—The following additional data of this steel sheathed cruiser, partially described on page 385 of Vol. VIII, are from the "Naval and Military Record," London.

The twin screw, vertical, four cylinder, triple expansion engines have cylinders 40, 60 and (2) 66 inches in diameter, the stroke being 39 inches. The I.H.P. under forced draft is expected to exceed 18,000, the corresponding speed being 23 knots. The normal coal supply, on a draught of 20 feet, will be 550 tons; the full supply that can be carried will be 1,000 tons.

The complement will be 500.

O'Higgins.—This steel sheathed cruiser is building by Messrs. Armstrong, Mitchell & Co., Newcastle-upon-Tyne. The "Naval and Military Record," London, gives the following particulars.

The length will be 412 feet; beam, 62.75 feet; displacement, 8,500 tons. The sheathing will be of teak. The armament will consist of four 8-inch guns, ten 6-inch, four 4.7-inch, ten 12-pounder and ten 6-pounder rapid fire guns, four Maxim machine guns and three torpedo tubes.

The twin screw engines are designed to develop 16,500 I.H.P. and give the vessel a speed of 18 knots. The boilers will be of the Belleville type.

Ministro Zenteno.—This steel cruiser was partially described on page 385 of Vol. VIII.

The twin screw, triple expansion engines are designed to develop 7,000 I.H.P., under forced draft, giving the vessel a speed of 20 knots.

Guardia Marina Riquelme and **Teniente Serrano.**—The third and the last of the thirty-knot destroyers built by Messrs. Laird Bros., Birkenhead, completed their official trials on December 18 and 17, 1896, respectively. For the purpose of comparing the four destroyers of this class, the description given on page 807 of Volume VIII, is here repeated, and the trial results tabulated below.

The length is 213 feet; beam, 21.5 feet; depth, 12.75 feet; displacement, about 300 tons. I.H.P., about 6,000. The armament consists of one 12-pounder and five 6-pounder rapid fire guns and two 18-inch torpedo tubes. The water tube boilers are of the Normand type, modified. There are four smoke pipes. The bunker capacity is 90 tons and the complement, 65 officers and men.

All four boats carried on the trials the contract load of 35 tons. The results were, as follows:

	Date of trial	Steam pressure.	Revolutions, on miles.	Speed, on miles.	Revolutions for 3 hours.	Speed for 3 hours.
Capitan Orella.....	Sept. 30	215	361	30.17	362	30.23
Capitan Munoz Gamero.....	Oct. 15	214	368	30.42	364	30.08
Teniente Serrano.....	Dec. 17	216	371	30.34	370	30.26
Guardia Marina Riquelme.	Dec. 18	215	362	30.09	362	30.12

The I.H.P. developed on the *Orella* was 6,200. After the full power trial of the *Riquelme*, the vessel was run over the mile, full speed astern, and attained a speed of 20.5 knots, a fairly straight course being steered during this time.

From the 8 hours' coal consumption trials of these vessels, at 13 knots, the radius of action at that speed was found to be about 3,750 nautical miles.

Torpedo Boats.—Two of the six first class boats, building by Messrs. Yarrow & Co., were launched on December 3, 1896, and January 16, respectively. The first one had her official trial on December 16. These boats were described on page 611 of Volume VIII; the data are repeated here.

They are 152.5 feet long and have a beam of 15.25 feet. The guaranteed speed with a load of 25 tons is 25.5 knots, and the designed I.H.P. of the triple expansion engines, about 2,000. The bunker capacity is 40 tons. The armament consists of three rapid fire guns and two torpedo tubes. These vessels will be an improvement on the Austrian torpedo boat *Viper*, built by Messrs. Yarrow & Co., and described on page 384 of Volume VIII.

The official trial was made in the estuary of the Thames, with the following results. Steam pressure, 175 pounds per square inch; revolutions per minute, 373; speed, for six runs over the mile, 26.807 knots, and for the three hours, 26.797 knots.

The remaining four boats are to be shipped in pieces and put together in Chili. The first one of these four has been tested

under steam, in the presence of Chilian officials, having been completely erected and bolted together in the contractor's yard.

CHINA.

Torpedo Cruisers.—Four of this type, building by Mr. F. Schichau, Elbing, are to be completed this year. The hulls will be of nickel steel. The I.H.P. is to be about 6,410 and the speed, 32 knots.

DENMARK.

Herluf Tralle.—This coast defense vessel is to be commenced during the present year at the dockyard at Copenhagen. The displacement will be about 4,920 tons.

ENGLAND.

Powerful.—On page 812 of Volume VIII, the first part of the official trials of this first class cruiser were described. On November 27, the final power trials, at 25,000 and 22,000 I.H.P. were satisfactorily completed. As was previously stated, the attempts at the full power run, October 17 and 21, were not successful. One boiler was put out of commission by the bursting of a pipe to one of the water gauge columns, and the heat in the uptakes was so great, owing to faulty firing, that water from the hose on the upper deck had to be used. The vessel returned to the Portsmouth dockyard, where the various defects were made good, and the smoke pipe casings lagged with four inches of silicate cotton, this lagging extending up to the upper deck. The smoke pipes were also lengthened ten feet, making a total height of ninety feet above the grates.

On November 26, these repairs and alterations were completed and a preliminary trial was satisfactorily made, the I.H.P. for one hour being over 25,000. On the next day, the final trials were made. The following description and data of the trials of November 27 are compiled from "Engineering" and "The Engineer," London.

In the *Powerful*, the contractors, after completing the four hours' run at full power, had to continue for the subsequent

four hours at 22,000 indicated horse power, so that for eight hours the boilers were subjected to a test of unusual severity. The result eminently justified the step taken in fitting the Belleville boilers in the *Powerful* and *Terrible*, and Mr. A. J. Durston, C. B., the Engineer-in-Chief, is to be congratulated on the courage shown in carrying through his policy amidst adverse criticism, as well as on the success attending it.

On the first four hours' run the mean power was 25,886 I.H.P., and for the greater part of the time steam was escaping from the exhaust pipe, so that there was abundance of steam. In the succeeding four hours the mean power was 22,634 I.H.P. Both these results were got under natural draft, the stokehold being in free communication with the atmosphere. All bulkhead and ash hoist doors were open, and the boiler casing covers were down, but not air tight. Before going into the details, it may be interesting here to compare the best results obtained with other types of naval vessels working with natural draft.

	Powerful.	*Blenheim.*	*Hawke.*	*Thetis.*
Weight in fire rooms, tons	1,164	754	628.5	402.3
I.H.P.	25,886	14,924	10,761	7,034
I.H.P. per ton of fire room weights	22.24	19.7	17.1	17.4
I.H.P. per sq. ft. G. S.	11.8	13.1	13.2	13.6
H.S. per I.H.P.	2.68	2.0	2.3	2.2

In the *Juno* class the power per unit of weight was 14.2 per I.H.P. The *Blenheim* got 28.3 I.H.P. per unit of weight in the boiler rooms, but under conditions of forced draft which, if not dangerous, were at least at the expense of the boilers, and will probably never be repeated. Thus in the latter cruisers prudence has dictated 20 I.H.P. per ton of boilers as the limit under forced draft, yet the *Powerful* gets over 22 under natural draft without any difficulty.

One point which developed itself in connection with the trials was a heating of some of the eccentric straps, owing partly to the fact that brass was adopted for lining, and partly to the great load on the valves consequent on the high steam pressures in use. In the *Powerful* the steam pressure in the I.P. receiver averaged 69 pounds. The substitution of white metal for the

brass, and one or two other slight adjustments, however, overcame the difficulty.

The *Powerful* left Portsmouth on Thursday, November 26, with 3,000 tons of coal on board.

Before daybreak on Friday, the vessel started westward to commence her runs over the measured course between Rame Head and Dodman Point, beginning with a run to the eastward.

Under ordinary conditions, indicator diagrams are taken every half hour, but as it was determined to run the ship three times over the course of between 22 and 23 nautical miles, involving a quick turn at either end, the half-hourly period might have occurred at the moment of turning, when the drag on either engine would have materially affected the result. It was therefore decided to take two sets of diagrams on each run—one a short time after the ship got on the course, and the other before the 22 miles were traversed. It was found at the end of the trial that the mean number of revolutions during the whole four hours, including the turnings, was only 0.2 of a revolution per minute below the mean on the straight portions of the course. The first and last runs were against the wind. With the wind which travelled at exactly the same speed as the ship, an increase in revolutions more than compensated in the power for the reduced pressure.

The boiler worked well throughout. The instructions were, "Fire light, fire often, fire quickly." It was the intention to fire each of the two fires in one boiler every three instead of every four minutes, as before, and to fire not more than four shovels full of coal at a time. It was found impossible, however, to carry this out, owing to the difficulty which an unaccustomed eye has in judging just what the thickness of the fire is and where the coal is most wanted. The fires were, therefore, kept from 5 to 7 inches thick and coal put on every four minutes only in sufficient quantities to fill up the holes or hollows.

A "devil's claw" was used on each fire about every quarter of an hour, for the purpose of levelling the fire. Fires were cleaned when necessary. The clinker was broken up with a slice bar in

the usual way and then quickly withdrawn, the fires being allowed to burn down slightly for the short time necessary.

It has been demonstrated that success is largely a matter of careful stoking, a light fire being conducive to economy and high evaporative efficiency, since it keeps the furnace in an incandescent state. After the coal consumption trials, the space between the firebars was increased by $\frac{1}{16}$ inch, making it $1\frac{3}{16}$ inch, while the bars are $\frac{3}{4}$ inch thick.

The draft plates were kept open about $\frac{3}{4}$ inch, the supply of air being adjusted by means of the twelve fans in the fire rooms. The ash pans were kept fully covered with water.

It may be noted that the primary purpose of these fans is for ventilating, and not for forcing the draft, as this is done by the Belleville air compressors, one in each fire room. These blow air over the fires through a pipe with eight nozzles, tapering from $\frac{3}{16}$ to $\frac{1}{8}$ inch. The mean air pressure thus obtained was 13.85 pounds per square inch. The ventilating fans are 6.5 feet in diameter and were run at a mean speed of 290 revolutions per minute, their designed speed being 300.

During the previous trials, the ashes were burned regularly in banks all along the sides of the grates. On the full power trial, where economy was not sought, this was not done. The draft was greatly improved by the lengthened smoke pipes, and it is expected that, in future, the time between firing can be increased to 5 minutes. Three of the four smoke pipes are oval in shape, each 10.75 feet in a fore-and-aft direction. The cross diameter of the forward one is 4.75, and of the next two, 7.5 feet. The after one is circular, 10.75 feet in diameter.

The service complement of petty officers, firemen and coal passers was on duty. Each watch consisted of the following men in each fire room.

1 petty officer in charge.
2 men for firing the four boilers.
2 men for breaking up coal, raking out ash pits and cooling ash pans.
1 man for supplying coal.
1 man to look after feed, air compressing and fan engines.

During the coal consumption trials, an additional man was in each fire room to tally coal.

The number of engineer officers was, of course, greater for the trials than the regular complement. For comparison, the regular complements of officers and men in the engineer's force are given.

	Chief engineer.	Engineers.	Artificers.	Chief stokers.	Leading stokers.	Stokers.
Blake	1	6	14	6	14	123
Powerful	1	7	18	9	27	219

For continuous or service steaming, when the men are in three watches, the *Powerful* is required to develop 18,000 and the *Blake*, 12,000 I.H.P. The maximum powers are 25,000 and 20,000 I.H.P., respectively.

The temperature of the air on deck was 52 degrees; in the ammunition passages, which run alongside the boiler casings under the protective deck, the highest record, after ten hours' steaming, was 82 degrees; in other places it ranged from 75 to 79 degrees. Before the lagging had been put on, a temperature of 140 degrees was registered. In the after fire room the temperature ranged from 80 to 97 degrees; in the middle rooms, from 78 to 96 degrees; and in the forward fire room, which is narrower than the others, from 75 to 110 degrees. These readings were taken at various places, but not immediately over the boiler casings under the protective deck. The temperature in the port engine room ranged from 61 to 87, and in the starboard room from 54 to 90 degrees.

In the H.P. cylinder during the full power trial the cut-off was 74, I.P., 78, and in the L.P. cylinders, 76 per cent. of the stroke. The main links were full out, but the H.P. independent link was $\frac{3}{8}$ inch in gear, while the others were not in use. The safety valves were loaded to 260 pounds per square inch. It may be noted that the port engine for a less number of revolutions gave a higher power than the starboard engine. This may have been due to a slight difference in the pitch of the propellers. For at least an hour the engines were developing over 26,000 I.H.P., and then

everything was working well. This was obtained with 116 revolutions; but the engines were for some time running at 119 revolutions, a very high speed for such massive machinery.

The following table gives the mean results of the four hours' 25,000 I.H.P. trial. The draught forward was 26.25 feet and aft, 28 feet and 1 inch; force of wind, 5 to 6; moderate sea. The coal consumption was not taken. The speed was obtained by landmarks.

Steam pressure in boilers, pounds per square inch	257.5	
	Starboard.	*Port.*
at reducing valve, pounds per square inch	227.6	230.3
engines, pounds per square inch	208	205
Revolutions per minute	115	113.8
Vacuum, inches	26.2	25.0
Mean pressures in cylinders, H.P.	93.2	96.9
I.P.	38.6	42.4
L.P., forward	17.5	17.5
L.P., aft	17.0	18.2
I.H.P., H.P.	4,131	4,256
I.P.	4,151	4,507
L.P., forward	2,218	2,190
L.P., aft	2,155	2,278
collective	12,655	13,231
total	25,886	
Vacuum in smoke boxes, inch	.33	
Speed, knots	21.8	

The speed of the ship, 21.8 knots, is for a mean of 114.9 revolutions and 25,886 I.H.P., but the weather conditions were not favorable to a speed test. Over the same measured course on a previous run the speed for 112.32 revolutions and 24,763 I.H.P. was 22.03 knots, so that, under similar conditions, 22.25 knots would easily have been attained. The details of the runs over the measured course on November 27, prove the unfavorable conditions. In the first run—west to east—against the wind, which was of force 6, and also against a rough sea, the speed was 21.451 knots, the I.H.P. being 25,733. The second run—east to west—was made with the wind and a following sea, which caused the ship to "yaw" considerably, making the steering of a straight course a matter of difficulty. The speed was only 21.945 knots, although the I.H.P. was 26,290. The wind trav-

elled at the same speed as the ship. In the third run, the speed was 21.746 knots, although the I.H.P. power was 25,635. The speed, therefore, is unreliable; but there is no doubt that 22.25 knots may be got with the power. The *Powerful* is to go through a series of progressive speed trials when she passes into the Steam Reserve, and then more reliable data may be got, as the primary consideration will not be a power test. Similar data are to be got for the *Terrible*.

There is an entire absence of vibration at the higher speeds of these vessels. The "beat" of the *Powerful* when developing 25,000 I.H.P. was not more perceptible than that of the pulse of a man of full vigor. No Atlantic liner is so free from vibration. The working at the "expansion joints" of the superstructure did not exceed $\frac{1}{16}$ inch.

Immediately after the four hours' full power trial, the *Powerful* continued at 22,000 I.H.P. for four hours. Coming after this trial, it was severe, but the same system was adopted below with equally satisfactory results. The coal was Harris' hand-picked Navigation; it burned well, and the fires had not to be clinkered much during the eight hours of the trial. The initial steam pressure was slightly reduced. The cut-off in the engines was varied somewhat as the power rose or fell, for the contractors were expected to make the mean within five per cent. of the stipulated total. For the greater part of the time, however, the cut-off in the H.P. cylinder was at 69, I.P., 72, and L.P. cylinders, 70 per cent. of the stroke. The mean results of the 22,000 H.P. trial are given in the following table. The speed was not taken.

Steam pressure in boilers, pounds per square inch	237.5	
	Starboard.	*Port.*
at reducing valve, pounds per square inch	220	216
engines, pounds per square inch	195	198
Revolutions per minute	109.6	109.5
Vacuum, inches	26.2	26.0
Mean effective pressure in cylinders, H.P.	87.1	90.3
I.P.	33.8	38.2
L.P., forward	16.1	16.0
L.P., aft	15.5	16.7
I.H.P., H.P.	3,684.	3,816.

	Starboard.	*Port.*
I.H.P., I.P.	3,464.	3,909.
L.P., forward	1,940.	1,935.
aft	1,867.	2,019.
collective	10.955.	11,679.
total	22,634	
Vacuum in smoke boxes, inch.	.34	

During this trial the vessel steamed back to Portsmouth. On November 28, she went out again in the teeth of a gale (force 7) to go through the manœuvring trials.

The time from full speed to stop for the starboard engine was 23 and for the port engine, 32 seconds. From stop to full speed astern, starboard, 12; port, 23 seconds; from full speed astern to full speed ahead, starboard, 11, and port, 15 seconds.

The steering engines, fitted to the engine room bulkhead, have two vertical cylinders, each 11 inches in diameter by 12 inches stroke, working through gearing. They are coupled by clutches to the one shaft, and so either can be engaged or disengaged. The rudder is of the usual balanced type common in the Navy, with an area of 270 square feet. Both steering engines were tried with the ship steaming about 18 knots. From midships to hard a-port, the time for the port engine was $12\frac{1}{2}$ and for the starboard engine, 11 seconds. From hard a-port to hard a-starboard, port engine, 26, and starboard, 21 seconds. Right back to hard a-port the time was 25 and 18 seconds, respectively; and, finally, from hard a-port to midships, 19 seconds for the port, and 9 seconds for the starboard steering engine. Trials were also made with the hand steering wheels, the steam gear being disconnected. The speed of the ship in this case was 19 knots, and 16 men with the three wheels put the rudder over 20 degrees to port in 2 minutes, bringing it back amidships in 40 seconds. Starboarding the helm 10 degrees took 10 seconds, and from 10 degrees to 15 degrees, 15 seconds; putting it over from amidships to 20 degrees occupied 56 seconds in all, and they brought the rudder back to amidships in 30 seconds.

Anchor trials were also made. Each anchor weighs $5\frac{3}{4}$ tons; the cables are $2\frac{9}{16}$ inches thick, each shackle of 75 feet in length weighing 2 tons. The engine working the heaving capstans has

two cylinders, each 14¼ inches in diameter by 12 inches stroke. Heaving and veering, with two bower anchors laid out, the speed was to be 25 feet per minute; heaving both, or veering both, or heaving one and veering the other, the speed was to be the same. Heaving in the slack cable, the speed was 40 feet. Catting with single chain, 20 feet, and heaving in slack cat chain, 40 feet per minute.

Terrible.—The trials of this first class cruiser, which were commenced on August 5, 1896, were successfully completed on January 9. Some results of the earlier trial were given on page 861 of Volume VIII. Since then new I.P. and L.P. slide valves have been fitted, the smoke pipes lengthened ten feet, and the smoke pipe casings lagged. On January 6, the last preliminary trial was made. Beginning with 70 revolutions per minute, these were increased to 106, and the engines then developed about 20,000 I.H.P. This was successfully maintained for two hours, the machinery working well. The following account of the trials and of the changes made in the valves is condensed from "Engineering," London.

The 30 hours' (5,000 I.H.P.) trial was made on August 5 and 6, the I.H.P. being 5,073, and the coal consumption 2.608 pounds per I.H.P. per hour. The average vacuum in the smoke box was .207 inch. The details are given in the following table.

Steam pressure in boilers, pounds per square inch	208.	
at engines, pounds per square inch	162.4	
	Starboard.	*Port.*
Vacuum, inches	26.6	27.0
Revolutions per minute	64.6	64.2
Mean pressure in cylinders, H.P., pounds	30.2	31.2
I.P., pounds	14.7	13.5
L.P., forward, pounds	7.	5.7
L.P., aft, pounds	7.5	5.9
Indicated horse power, H.P.	754.	772.
I.P.	887.	810.
L.P., forward	499.	406.
L.P., aft	530.	414.
Total indicated horse power	2,670.	2,402.
Collective indicated horse power	5,073.	
Speed, from four runs over the mile, knots	13.43	
Coal per I.H.P. per hour, pounds	2.608	

Runs were made on the measured mile, but as some part of the power was lost in undue cylinder friction, for a reason to be referred to later, the speed realized is not reliable; the vessel will be put on a speed trial at low powers now that the engines have been readjusted; and this readjustment forms an interesting story.

The H.P. cylinder has a piston valve, the I.P. and the two L.P. cylinders have slide valves. The H.P. cylinder is forward, with its piston valve on the after side; the I.P. cylinder comes next, with its slide valve abaft; while the valves of the two L.P. cylinders are both placed between the cylinders. The H.P. and I.P. cranks are at right angles, and the two L.P. cranks are at right angles to each other. The high steam pressure, 160 pounds per square inch on the H.P. piston, brought prominently forward the whole question of the load on the back of slide valves, and some interesting problems have developed. A special type of relief ring was fitted to the back of the valves, and the cut-offs were originally arranged by the contractors so that the powers developed by the cylinders would be equal. The details of the 5,000-H.P. trial show that this equalization was realized. But when the 18,000-H.P. trial came to be run, it was found that the pressure on the I.P. valves was too great, and the material of the face being soft, it wore off rapidly, involving the construction of new valves.

The new valves have harder faces, this being preferred only because of the high pressure, although softer metal has a less tendency to crack. Each valve is 6 feet 4 inches square, giving an area of 40.1 square feet. It is stiffened by ribs on the bars, and the face at the back has been chamfered off to ease the working. The relief ring is 4 feet $\frac{3}{4}$ inch in diameter. When running to get the ordinary ratios of work from the respective cylinders for a total of 18,000 I.H.P., the pressure in the first receiver of each engine was 92 pounds, equal to a load of 76 tons on the valve when the relief ring was inoperative. It was, therefore, decided to reduce this pressure.

The valves were altered, $\frac{3}{8}$ inch being taken off the steam lap and $\frac{1}{4}$ inch from the exhaust lap. The eccentrics were also

altered to reduce the receiver pressures; the I.P. was put back in the shaft to give a maximum pressure of about 70 pounds, while the L.P. was put back to decrease it from 30 to 25 pounds. In addition to the ordinary all-around reversing gear, by-pass valves were fitted to admit steam to the I.P. valve casing or second receiver. Before this there was a starting valve which admitted steam direct to the cylinders, but this had a tendency to blow the slide valves off the faces. The 5,000-H.P. trial was made with the original setting of the valves. The subsequent trials were made with the new arrangement, and the vacuum at the back of the relief rings was 17 inches for the I.P. and 21 inches for the L.P. valves.

When the trouble with the valves was first experienced the cut-off in the H.P. cylinder was at 59, I.P., 62, and L.P. cylinders, 50 per cent. of the stroke. On the trial of January 7 and 8, the H.P. cut-off remained at 59 per cent., but was changed to 74 and 64 for the I.P. and L.P. cylinders, respectively. On the full power trial, the cut-offs in the starboard engine were, H.P., .743, I.P., .741, forward L.P., .593, after L.P., .624; and in the port engine, H.P., .733, I.P., .747, forward L.P., .561, and after L.P., .63. In the 22,000-H.P. trial the cut-off was a little earlier in all cases.

The 18,000-H.P. trial, commenced on Thursday, January 7, should have been of thirty hours' duration; but the weather was thick and stormy, and it was deemed desirable to anchor inside the breakwater before dark, so that it was decided that a twenty-nine hours' trial would be sufficient.

The number of revolutions varied between 101 and 105 per minute, and the I.H.P. between 18,016 and 19,083. The mean results for the 29 hours are given in the following table.

COAL CONSUMPTION TRIAL, JANUARY 7 AND 8, 1897.

Draught, forward, feet and inches	26– 1	
aft, feet and inches	28–10	
Steam pressure in boilers, pounds per square inch	223.5	
	Starboard.	*Port.*
Steam pressure at engines, pounds per square inch	202.	196.
Vacuum, inches	25.8	26.5
Revolutions per minute	102.4	103.

	Starboard.	*Port.*
Mean pressure in cylinders, H.P.	70.78	77.3
I.P.	28.62	29.41
L.P., forward	15.61	15.31
L.P., aft	16.19	14.59
I.H.P., H.P.	2,844.	3,106.
I.P.	2,738.	2,827.
L.P., forward	1,761.	1,736.
L.P., aft	1,826.	1,655.
total	9,169.	9,324.
collective	18,493.	
Coal per I.H.P., mean for 29 hours	1.71	
Vacuum in smoke boxes, inch	.31	
Speed, by log, knots	20.964	

The *Powerful* developed on the corresponding trial 18,433 I.H.P. for a mean of 102.8 revolutions—practically the same result—and the speed was the same. The coal efficiency was greater in the *Terrible;* the boilers had their fire bars slightly closer. The pressure of the air supplied through the $\frac{3}{16}$-inch nozzles over the fire was 9 pounds to the square inch. The vacuum in the smoke box was measured at the peep holes in the front of the boilers. At no time during the trial was the coal consumption over 1.9 pounds per I.H.P., and for five hours it was below 1.6 pounds, seven times more under 1.7 pounds, fourteen times under 1.8 pounds, and only three times over 1.8 pounds. No fires were cleaned until the eighth hour, and then the clinker was removed at long intervals. Harris' Deep Navigation coal was used. The consumption was equal to 14.1 tons per hour, which, for a speed of practically 21 knots, is satisfactory, and will compare most favorably with Atlantic liner performances. This, too, makes no allowance for auxiliary machinery. In the United States, French and other navies, such deduction is made; but even if the power developed by these be accurately indicated from time to time, there is difficulty in estimating the coal used, as they are of various types to suit special requirements—simple, compound, or triple. On the *Terrible* there were in use throughout the 29 hours, six main feed pumps, four main circulating pumps, two auxiliary circulating pumps, two hotwell pumps, four fire and bilge pumps, two engine room fans, one drain tank engine, twelve air pumping engines, two of

the three sets of electric light engines (working at nearly full load and developing about 180 I.H.P.), and twelve fans. Steam was always on the steering engine, and two distiller pumps were used for 26 hours, the fan in the evaporator room for 20 hours, and twelve ash hoists were used ten minutes in each watch of four hours. Thus the 14.1 tons per hour is enough for all service requirements, and as the vessel may carry 3,000 tons, she could travel 4,000 nautical miles at this speed.

The coal burned per square foot of grate area was 14.5 pounds, and at this rate the evaporation per pound of coal was about 10 pounds of water from 100 degrees. This was reckoned from the strokes of the feed pumps; as a matter of fact they showed 11.4 pounds; but deduction is made for slip.

On Saturday, January 9, the four hours' full power trial was made, about 14 hours after the previous trial, leaving this much time to overhaul engines, sweep tubes, and prepare for the trial. It was arranged to run three times over the course between Dodman Point and Rame Head. The tide at this course, even at the flood, is not over ½ knot. Indicator cards and observations were taken near the beginning and end of each run. The first run was against tide and wind, the force of the latter being from 6 to 7. On the third run the wind had freshened.

When the ship was going with the wind, the latter was traveling at the same speed as the ship, and the fans were then run at 354 revolutions per minute. The stokehold covers at the base of the funnel were closed; but there was free communication between the stokeholds. The pressure in the stokeholds was, perhaps, slightly above the atmosphere; there was an appreciable draft into the engine room, and a canvas screen was loosely put up to keep dust out. These facts show that it was practically a natural draft trial; the vacuum, taken as before mentioned, was .32 inch, and the 22.25 I.H.P. per ton in the boiler room is consequently a very good result. The air compressing engines were worked at 103 revolutions, the pressure being 14.6 pounds to the square inch.

With the exception of the fourth set of results, when the revolutions were 114, the engines did not vary much from 111 or

112 revolutions. The total power was gradually worked up to 26,479 I.H.P. with 114 revolutions, then decreased, both engines running at the same speed. The variation was .1 revolution per minute, so that they proved very sensitive to adjustment. The effect of the new arrangement of valves will be seen by reference to the steam pressures in cylinders, and the I.H.P. of the separate cylinders. The mean results are given in the following table.

FULL POWER TRIAL, JANUARY 9, 1897.

Draught, forward, feet and inches	25–9	
aft, feet and inches	28–3	
Steam pressure in boilers, pounds per square inch	229 6	
	Starboard.	*Port.*
Steam pressure at engines, pounds per square inch	201.	194.
Vacuum, inches	26.	25.9
Revolutions per minute	112.	111.98
Mean pressure in cylinders, H.P.	82.	91.70
I.P.	33.41	37.20
L.P., forward	21.60	20.85
L.P., aft	22.41	21.10
I.H.P., H.P.	3,601.	4,005.
I.P.	3,491.	3.885.
L.P., forward	2,661.	2,568.
L.P., aft	2,762.	2,599.
total	12,515.	13,057.
collective	25,572.	
Speed, knots	22.41	

The maximum temperature recorded in the engine rooms was 95, starboard, and 109 degrees, port, the temperature on the forecastle deck being 47 degrees. The temperature was also taken immediately over the boilers in the stokehold; only in the after stokehold was a record obtainable—248 degrees; in all other cases the thermometers burst. Again the temperature was taken on the level of the upper deck, at the base of each of the funnels, but inside the funnel casing. Starting forward, the readings were 192, 216, 224 and 138 degrees for the four funnels. The ammunition passages are close to the uptake casings, and the readings in the upper passages were 98 to 105, and in the lower, 74 to 82 degrees. With all the doors closed, the highest reading was 112 degrees, which is not abnormal, even for cordite.

Immediately following the full power trial came the four hours' trial at 22,000 I.H.P. The speed of the engines was reduced to 108.83 revolutions, which gave 22,282 I.H.P., while the means for the same trial of the *Powerful* were 109.5 revolutions and 22,634 I.H.P.

The mean results are given in the following table.

22,000-H.P. TRIAL, JANUARY 9, 1897.

Draught, forward, feet and inches	25–9	
aft, feet and inches	28–3	
Steam pressure in boilers, pounds per square inch	225.	
	Starboard.	*Port.*
at engines, pounds per square inch	200.	193.
Vacuum, inches	26.1	25.84
Revolutions per minute	108.8	108.86
Steam pressure in cylinders, H.P.	79.30	80.51
I.P.	30.	34.01
L.P., forward	18.62	18.16
L.P., aft	19.23	18.86
I.H.P., H.P.	3,382.	3,419.
I.P.	3,047.	3,466.
L.P., forward	2,229.	2,186.
L.P., aft	2,304.	2,249.
total	10,962.	11,320.
collective	22,282.	
Vacuum in smoke boxes, inch	.34	

Towards the end of the above trial, the stokers somewhat relaxed their vigilance of the system and care which had been enforced during the three days; they allowed their fires to get thick, and, as a result, a great torch was lighted on top of three of the funnels. It was proof of the efficiency of stoking in the earlier parts of the trial that nothing of the kind then took place.

A comparison of the speeds of the *Powerful* and *Terrible* on the 18,000 and 25,000-I.H.P. trials will be of interest, owing to the difference in the propellers. The *Powerful's* propellers, which are of Admiralty gun metal, were suggested by experiments in the Government experimental tank at Haslar; while the *Terrible's* propellers are those suggested by the long experience of Atlantic work of the contractors. The diameters of the propellers of the two vessels are the same, 19.5 feet. The *Terrible's* propellers,

which are of Parsons' manganese bronze, have a mean pitch of 24 feet, and each has 92 square feet of surface, against 73 square feet in the case of the *Powerful*. The blades of the *Terrible's* propellers were polished.

The following table gives the means of the data of three runs over the 23-knot course.

	I.H.P.	Revolutions.	Speed.	Slip.
18,000-H.P. trial.				
Powerful	18,678	103.2	20.96	13.7 (?)
Terrible	18.522	103.53	20.96	14.5
25,000-H.P. trial.				
Powerful	25,886	114.4	21.8	18.5
Terrible	25,572	112.0	22.41	15.2

For the full power trials, the force of the wind was about the same, 6 to 7; the sea perhaps was more adverse for the *Powerful*, causing the ship to "yaw" during one of the runs. The mean slip is worked out independently for the mean of mean speeds.

The draught of the *Terrible*, given in the preceding tables, corresponds to the designed displacement, 14,200 tons.

Two runs of the progressive speed trials have been made on the measured mile in Stokes Bay, with the following results. A 1-knot tide was running, but other conditions were favorable.

With the tide, speed, 11.25 knots; revolutions, starboard, 49.12, port, 47.62; I.H.P., starboard, 1,115, port, 893. This gives a mean of 48.8 revolutions, and a total of 2,008 I.H.P. Against the tide, speed, 9.326 knots; revolutions, starboard, 50.51, port, 48.03; I.H.P., starboard, 1,180, port, 907. The mean was 49.27 revolutions, and the total 2,087 I.H.P. For the two runs the means were, speed, 10.288 knots; revolutions, 49; I.H.P., 2,047.

Juno.—This sheathed, second class cruiser, built by the Naval Construction and Armaments Co., Barrow-in-Furness, and described on page 169 of Vol. VIII, completed satisfactory official trials on November 10, 1896. The vessel was laid down on June 22, 1894, launched on November 16, 1895, and brought to the Devonport dockyard in August, 1896, to be docked and made

ready for the trials. While in dock, the pitch of the propellers was set at 16 feet 7.5 inches. The total cost of the vessel, ready for sea, is given in the Navy Estimates as $1,356,485, of which the contractors get $787,125, for the hull, fittings and equipment, and $275,025, for the propelling and other machinery, not including steamboats; the amount for guns is $86,000; the balance is taken up for dockyard work to complete the vessel.

The estimated I.H.P. and speed, under forced draft, were 9,600 and 19.5 knots; under natural draft, 8,000 and 18.5 knots, respectively.

The first of the official trials, 8 hours' natural draft, was delayed, by the discovery of defective safety valves, until October 26, when satisfactory results were obtained as follows: Steam pressure in boilers, 153, at engines, 151 pounds per square inch; vacuum, starboard, 24.8, port, 27.1 inches; revolutions per minute, starboard, 138.6, port, 138.3; I.H.P., starboard, 4,116, port, 4,156, total, 8,272; air pressure, .42 inch; speed, 18.8 knots.

On October 28, a four hours' forced trial was made, but the I.H.P. was not developed, although a speed of 19.5 knots was attained. Another trial was ordered to take place after the coal consumption trial had been made. Some adjustments having been made, the forced draft trial was successfully completed on November 10, the results being as follows.

Steam pressure in boilers, 154, at engines, 151 pounds per square inch; vacuum, starboard, 26, port, 26.6 inches; revolutions per minute, both engines, 149.3; I.H.P., starboard, 4,832, port, 4,939, total, 9,771; air pressure, .92 inch; speed, 20 knots, or ½ knot above the contract. Draught of vessel, forward, 19 feet 11 inches, aft, 21 feet 7 inches, mean, 20.75 feet; displacement, about 5,600 tons.

The 30 hours' coal consumption trial was finished on November 1, with the following results: Steam pressure in boilers, 142 pounds per square inch; vacuum, starboard, 26.1, port, 27.1 inches; revolutions per minute, starboard, 119, port, 117.9; I.H.P., 4,863; speed, by log, 16.3 knots; coal per I.H.P. per hour, 1.64 pounds.

The masts are to be shortened from 160 to 140 feet.

Doris.—This sheathed, second class cruiser, a sister vessel of the *Juno*, described above, has completed part of her official trials. The vessel was launched on March 3, 1896.

The eight hours' natural draft trial was made on December 5, with the following results. Steam pressure in boilers, 152, at engines, 150 pounds per square inch; vacuum, starboard, 27, port, 26 inches; revolutions per minute, starboard, 140.1, port, 139.7; I.H.P., starboard, H.P. cylinder, 1,360, I.P., 1,315, L.P., 1,594; I.H.P., port, H.P. cylinder, 1,318, I.P., 1,303, L.P., 1,502; I.H.P., collective, starboard, 4,269, port, 4,123; I.H.P., total, 8,392; speed, 19.1 knots.

The four hours' forced draft trial was made on December 7. Steam pressure at boilers, 154, at engines, 150 pounds per square inch; vacuum, starboard, 27, port, 26.25; revolutions per minute, starboard, 149, port, 148; I.H.P., starboard, H.P. cylinder, 1,493, I.P., 1,438, L.P., 2,046; I.H.P., port, H.P. cylinder, 1,465, I.P., 1,428, L.P., 1,981; I.H.P., collective, starboard, 4,977, port, 4,874; I.H.P., total, 9,851; air pressure, from 1.1 to 1.2 inches of water. Owing to the high seas and strong wind, the speed could not be accurately determined, although the runs over the measured course between Dodman Point and Rame Head showed an average of about 20 knots.

The indicator cards, taken half-hourly, showed that the revolutions per minute for the starboard engine varied from 148.7 to 151.1, the corresponding I.H.P. being 4,919 and 5,098; and for the port engine, from 147.7 to 150.5, the I.H.P. being 4,816 and 5,029. The maximum I.H.P. at any one time was, therefore, 10,127, with 150.8 revolutions.

After completing the four hours' trial, the turning trials were commenced, but were interrupted by the breakdown of the steering gear, the rudder shaft being bent. Repairs to this were completed on December 23. The masts are to be shortened from 160 to 140 feet.

Dido.—This sheathed, second class cruiser, described on page 388 of Volume VIII, completed her official trials on December 20, 1896. This vessel was laid down on August 30, 1894, launched on March 20, 1896, and was turned over by the contractors, the

London & Glasgow Shipbuilding and Engineering Co., Glasgow, on November 19, 1896, at the Chatham dockyard, there to be completed for the trials. The following additional particulars and the data of the trials are from "Engineering," London.

The total weight of the machinery in working trim is 900 tons. The eight single-ended boilers have a total grate surface of 610, and heating surface of 19,900 square feet. The contract required that 8,000 I.H.P. should be developed with not over .5 inch air pressure, the speed expected being 18.5 knots, and with forced draft, 9,600 I.H.P., the speed expected being 19.5 knots.

On December 16, the eight hours' natural draft trial was made with the following results: Draught of vessel at beginning of trial, forward, 19.75 feet, aft, 21.75 feet, mean, 20.75 feet; steam pressure, 150 pounds per square inch; vacuum, starboard, 26.9, port, 25.6 inches; revolutions per minute, starboard, 136, port, 137.4; I.H.P., starboard, 4,111, port, 4,228, total, 8,339; air pressure, .19 inch; speed, by patent log, 19 knots. Weather fine.

On December 18, the four hours' trial under forced draft gave the following results: Steam pressure, 150 pounds per square inch; vacuum, starboard, 26.7, port, 26 inches; revolutions per minute, starboard, 143.6, port, 144.7; I.H.P., starboard, H.P., 1,596, I.P., 1,632, L.P., 1,676; I.H.P., port, H.P., 1,648, I.P., 1,560, L.P., 1,751; collective I.H.P., starboard, 4,904, port, 4,959; I.H.P., total, 9,863; speed, by patent log, 21.1 knots.

The highest power developed during the trial was 10,300 I.H.P., with a mean number of 147.2 revolutions per minute; the lowest, 9,592 with 143.7 revolutions per minute.

On December 20, the 30 hours' coal consumption trial at half power was completed. All of the eight boilers were used. Draught, forward, 19.5, aft, 21.5 feet. The results were as follows: Steam pressure, 147 pounds per square inch; vacuum, starboard, 26, port, 27.5 inches; revolutions per minute, starboard, 117.3, port, 117.1; I.H.P., starboard, 2,473, port, 2,452, total, 4,925; speed, by log, 17.7 knots; coal per I.H.P. per hour, 1.6 pounds.

Furious.—This second class cruiser was launched at the Devonport dockyard on December 3, 1896. She is a sister vessel of the *Arrogant*, laid down at the same yard at the same time, and des-

cribed on page 617 of Vol. VIII. The principal dimensions and data are repeated, and some particulars, from "Engineering," London, added.

Length, 320 feet; breadth, 57 feet 6 inches; mean load draught, 21 feet; displacement, 5,800 tons. Her keel was laid on June 10, 1895. The armament will consist of rapid fire guns entirely, there being four 6-inch, six 4.7-inch, and twelve smaller guns, besides five Maxim machine guns. There are two 18-inch submerged torpedo tubes.

The twin screw engines, in separate compartments, are of the usual vertical, inverted, triple expansion type, and are to develop 10,000 I.H.P. under natural draft, and give the vessel a speed of from 18.5 to 19 knots. The cylinders are 26, 42 and 68 inches in diameter, the stroke being 39 inches.

The total cooling surface is 5,280 square feet. The machinery was built by the Earle Shipbuilding Co., Hull, who also built that for the *Arrogant*.

The eighteen Belleville boilers, some particulars of which were given on page 170 of Volume VIII, under the sister ship *Vindictive*, have a grate surface which is now given as 860 square feet, and a heating surface of about 23,000 square feet. As the boilers are placed in three equal groups in separate compartments, there will be three smoke pipes.

One hundred feet of the length of the ship is given over to the eighteen boilers and the coal supply, and 48 feet to the machinery, although this includes much of the auxiliary machinery, *i. e.*, dynamos, etc.

The complement of officers and men will be 419.

The weight of the hull is 2,655 tons, and the launching weight was between 2,700 and 2,750 tons.

The coal capacity at load draught is 500 tons. There are bunkers above and below the protective deck.

Special attention has been devoted to the strengthening of the ship forward, so that she may use her ram without the least fear of serious damage resulting to the structure. The other vessels of this class, *Gladiator*, *Vindictive* and *Arrogant*, are similarly strengthened. The stem is massive and well braced to the inte-

rior structure, the framing is heavy, the shell plating is 1 inch thick, and over this there is nickel steel armor plating, 2½ inches thick, extending for a very considerable distance aft, although with decreasing depth. At the stem it is the full depth of the ship, but the top edge curves downwards and the bottom edge upwards, so that the belt continues at the water line for a distance of quite 50 feet from the ram.

Effective speed and manœuvring will greatly assist calm judgment when ramming, and everything that wide experience can do has been done in the *Furious* class. There are two rudders, one forward, the other abaft, the twin propellers. The deadwood is cut away; in other words, the flat plate keel slopes upward, giving the propellers a clear sea in which to work. The after rudder is of the usual balanced type and worked in the ordinary way, the bottom of the rudder, like the propeller brackets, being supported on framing. The small auxiliary rudder forward is of the same type. It is worked by worm gearing from the same head as the main rudder.

The masts are to be shortened from 160 to 140 feet, and the vessel will be provided with the new "splinter netting." This netting will be of 4-inch mesh and fitted over and around the whole vessel and high enough to clear the heads of the men at the guns on deck.

Gladiator.—This second class cruiser, a sister vessel of the *Furious* described above, was successfully launched at the Portsmouth dockyard on December 8, 1896. It is reported that oil fuel will be used on this vessel.

Hermes, Highflyer and **Hyacinth.**—By the placing of the orders for these second class cruisers, the shipbuilding program for 1896–97, mentioned on page 376 of Volume VIII, is completed. The first two of these vessels will be built by the Fairfield Shipbuilding Co., Glasgow, and the *Hyacinth* by the London and Glasgow Shipbuilding and Engineering Co.

The two Fairfield vessels will be similar to the *Venus*, built by this company and described on page 830 of Volume VIII, and the other one will be similar to the *Dido*, described on page 388 of Volume VIII. The machinery of the new vessels will,

however, be different from that of the older vessels. The following particulars are from the "Marine Engineer" and "Engineering," London.

The twin screw, four cylinder, triple expansion engines will have cylinders 26, 42 and (2) 48 inches in diameter, the stroke being 30 inches. The number of revolutions per minute, for full power, is to be 180, and the working pressure at the engines, 250 pounds per square inch. The propellers will be three-bladed.

There will be eighteen Belleville boilers of the new type placed in three compartments, and increasing the number of smoke pipes from two, in the older vessels, to three. The total grate surface will be 750, and the heating surface about 24,000 square feet. The working pressure in boilers will be 300 pounds per square inch. The new type of Belleville boiler consists of two series of elements, one above the other, with a combustion chamber between them. The gases of combustion are broken up by air nozzles in the combustion chamber. The feed water, which passes through the upper series of tubes, is, therefore, heated to a high temperature before it goes to the lower series of tubes or those over the fire. This arrangement is said to increase the efficiency by 25 per cent.

Each boiler will contain nine elements, each element having seven pairs of 4.5-inch tubes. Over these will be the combustion chamber, and over the combustion chamber, nine more elements, each with seven pairs of 2.75-inch tubes, the latter forming the economizer. Thus each boiler will have 126 large and 126 small tubes, all of steel, solid-drawn. The weight of machinery will be 870 tons.

The armament will consist of eleven 6-inch and eight 12-pounder rapid fire guns; six machine guns, and three under water torpedo tubes.

The contract price of each vessel is reported as about $1,180,000, and they are to be finished in about two years.

Pelorus.—This third class cruiser, the first of a new type, which was launched at the Sheerness dockyard on February 15, 1896, and described on page 390 of Vol. VIII, completed satisfactory official trials on November 2, 1896. The following data

of the vessel and her trials are from "Engineering," London, except where otherwise mentioned.

The length is 300 feet; beam, 36.5 feet; draught, 13.5 feet; corresponding displacement, 2,135 tons. The weight of the hull is 1,110 tons. The twin screw, triple expansion engines have cylinders 20.5, 33 and 54 inches in diameter, the stroke being 27 inches. The eight Normand water tube boilers have a total grate surface of 350, and a heating surface of 15,800 square feet. The propellers are 11 feet in diameter. The contract I.H.P. was 7,000, and the expected speed, 20 knots.

She had exceptionally tall masts with two cross yards on each, besides the usual gaffs, but after the trials these masts were shortened 12 feet.

All trials were made with the vessel at her load draught of 13.5 feet.

According to the "Naval and Military Record," London, the vessel started on her eight hours' natural draft trial on October 6, but the trial was stopped by defective steering gear. On October 7, another trial was made, but, owing to leaky condensers, was not finished.

On October 8 or 9, the natural draft trial was successfully finished, with the following mean results: Steam pressure, 250 pounds per square inch; vacuum, 25.3 inches; revolutions per minute, 189; I.H.P., starboard, 2,668; port, 2,581; total, 5,249.

On October 10, according to the "Naval and Military Record," the 30 hours' coal consumption trial was commenced, but was stopped at the end of about 24 hours on account of priming in boilers.

This trial was successfully finished on October 17, with the following results. Steam pressure, 248 pounds per square inch; vacuum, 26 inches; revolutions per minute, starboard, 160, port, 162; I.H.P., starboard, 1,726, port, 1,841, total, 3,657; speed, by log, 16.25 knots. The coal per I.H.P. per hour is not given.

The forced draft trial of four hours' duration was commenced on October 20, but was, according to "Industries and Iron," London, unsatisfactory, the vessel returning to Devonport to have her propellers altered. Another trial, commenced on October 24,

was interrupted by the breaking of a bolt on the port engine. On October 27, another trial was interrupted by the breaking of a bolt of the drag link on the starboard engine.

On November 2, this trial was satisfactorily completed with the following results. Steam pressure, 260 pounds per square inch; vacuum, 25 inches; revolutions per minute, starboard, 215, port, 219 (?); I.H.P., starboard, 3,529, port, 3,499, total, 7,028. The speed, for three runs over the deep water measured course between Dodman Point and Rame Head, was 20.6, 20.7 and 21 knots, giving a mean of 20.73 knots. This speed makes the *Pelorus* the fastest vessel of her size in the British Navy. The weather was fine and the engines and boilers worked well, the steam pressure being easily maintained.

The weight of the machinery, with water, is given as 360 tons, so that the I.H.P. (7,028) per ton of machinery is 19.52. The heating surface per I.H.P. is 2.25, and the I.H.P. per square foot of grate surface, 20.08.

The machinery was built by Messrs. J. & G. Thomson.

Proserpine.—This third class cruiser, described on page 419 of Volume VII, and which is a sister vessel of the *Pelorus*, described above, was successfully launched at the Sheerness dockyard on Saturday, December 5, 1896. The vessel was laid down on March 2, 1895. The two vessels differ only in their boilers, the *Proserpine* having Thornycroft, and the *Pelorus*, Normand boilers. The complement will consist of 225 officers and men.

Pactolus.—This third class cruiser, built by Sir W. G. Armstrong & Co., Newcastle, and described on pages 391 and 834 of Volume VIII, was successfully launched on Monday, December 21, 1896.

Spitfire and **Swordfish.**—The trials and some particulars of these torpedo boat destroyers were given on pages 618 and 843 of Volume VIII. The following additional decription of the machinery and trials is from "Engineering," London, the cylinder dimensions being repeated for reference.

The twin screw, four cylinder, triple expansion engines, and nearly all the auxiliary engines were built by Messrs. G. E. Belliss

& Co., Birmingham. The main engines were designed to develop 4,000 I.H.P., with 400 revolutions per minute and a steam pressure of 200 pounds per square inch. The cylinders are 18.75, 27.25 and (2) 28 inches in diameter, the stroke being 18 inches.

The most noticeable point is the arrangement of valves, which presents some novel features. The H.P. and I.P. cranks are set opposite to each other, whilst the two L.P. cranks are opposite to each other, and at right angles to the two former. The H.P. and I.P. valves, and the two L.P. valves, are respectively combined in pairs, the two valves of each pair having a common valve rod, and being placed one above the other in one chamber. This arrangement, which is only permissible when the two cranks are placed opposite to each other, tends to a reduction of the number of moving parts, and enables the engine to be got into less fore-and-aft space. There is more room for main bearings, and there is a further advantage of a shorter weigh shaft, thus leaving the ends of the engine clear for the thrust block and air pump. This arrangement of valve gear has been used for some time past by Messrs. Belliss in their electric light engines, the satisfactory performance of which, both in regard to economy and effective working, led them to use the same arrangement in these vessels.

The air pump is worked direct from the main crank shaft, which is prolonged by means of a small crank shaft, the two being joined by a coupling. The arrangement, largely adopted in vessels of this class, enables the air pump to be placed so that it does not prevent access to the engine, the small fore-and-aft space it occupies not being of great importance. It is also very accessible. The diameter of the pump is 19 inches and the length of stroke, 4 inches. The pistons are of forged steel, and the top cylinder covers of steel, pressed to shape in dies by means of a hydraulic press. Each main engine has a cylindrical copper condenser with 929 $\frac{5}{8}$-inch tubes, the total cooling surface being 3,040 square feet.

The auxiliary machinery built by Messrs. Belliss, includes two starting engines, two centrifugal circulating engines, two air compressors, four fan engines, and one dynamo engine for each vessel.

There are eight Yarrow boilers in each vessel, the total grate surface being 178, and the heating surface, 8,500 square feet. The tubes are of steel 1 inch in diameter. The weight of the boilers without water, but complete with all fittings, is 41.2 tons per boat, giving a total output of 5.54 I.H.P. per hundredweight of boiler, based on the trial power (4,570) of the *Swordfish*.

Each boiler has an independent feed pump of the duplex type, with an additional pump in each of the two stokeholds for auxiliary feed. For forced draft there are four fans, each 4 feet 6 inches in diameter. They are placed horizontally, and are driven direct by double-acting engines.

The weight of propelling machinery, based on the 4,570 I.H.P. developed on the *Swordfish*, is at the rate of 46.63 pounds per I.H.P., made up by the following items:

	Pounds per I.H.P.
Weight of stern tube shafting and propellers, ...	2.35
Weight in engine room, including main engines, circulating engines, and other auxiliaries, condensers and all pipes and fittings, ...	17.42
Weight of boilers and water, funnels, casings, pipes and fittings in boiler rooms, ...	26.86
Total weight in pounds per I.H.P. developed, . .	46.63

The following results were obtained on the final trials of these vessels, the means being for the three hours:

	Spitfire.		*Swordfish.*	
Date of Trial	Aug. 17, 1896.		Oct. 21, 1896.	
Draught, forward, feet and inches	5–2.		5–2½.	
Draught, aft, feet and inches	7–4¾.		7–5.	
Steam pressure in boilers, lbs. pr. sq. in.	195.		203.	
	Strbd.	*Port.*	*Strbd.*	*Port.*
Vacuum, inches	24.3	23.6	24.1	24.4
Revolutions per minute	401.9	395.7	399.2	398.2
Mean pressure in cylinders, H.P.	68.	70.2	66.2	66.7
I.P.	30.7	33.3	36.4	36.2
L.P., for'd.	20.2	20.6	19.3	19.5
L.P., aft.	19.8	18.8	18.9	19.0
I.H.P., H.P.	686.	697.	672.	662.
I.P.	651.	700.	771.	764.

	Spitfire.		*Swordfish.*	
I.H.P., L.P., forward	453.	457.	431.	428.
L.P., aft	446.	416.	424.	418.
total	2,236.	2,270.	2,298.	2,272.
collective	4,506.		4,570.	
Speed, knots	27.461		27.117	
Air pressure, inches	3.09		2.3	

A trial to test fuel economy was made with the *Spitfire* on July 9, 1896. The machinery was worked at about one-tenth of the full power, when a speed of approximately 13 knots was made. At this rate, 33.2 nautical miles were covered per ton of coal burnt, and the fuel consumption was 1.97 pounds per I.H.P. per hour.

Virago.—This 30-knot destroyer, built by Messrs. Laird Bros., Birkenhead, and launched on November 19, 1895, completed satisfactory trials on November 27, 1896.

The displacement is 300 tons, and the designed I.H.P., 6,000.

In the early part of November, the first preliminary runs, under full power, were made over the Skelmorlie measured mile. The mean steam pressure was 216 pounds per square inch, and the speed, 30.508 knots. About a week after this, on November 12, a successful official, full power, coal consumption trial was made on the Clyde in the presence of the Admiralty representatives. The mean speed for six runs over the mile was 30.17 knots, and 30.07 knots for the three hours' continuous steaming. The coal consumption was 2.42 pounds per I.H.P. per hour. The mean steam pressure was 208 pounds per square inch, and the I.H.P., for the three hours, 6,425. The lowest speed made over the mile was 29.75, and the highest, 30.51 knots.

On November 27, the full power speed trial was successfully finished, the mean speed for the three hours being 30.049 knots, and the revolutions per minute, 363.05. The mean speed for six runs over the measured mile is given as 30.365 and 30.57, the number of revolutions per minute for the former speed being given as 364. The speed, 30.57 knots, is deduced as the regular third mean from the speeds on the separate runs as published. The steam pressure averaged 217 pounds per square inch. The weather outside the Cumbraes was not of the most favorable

description for a high result, the sea occasionally being choppy, and clouds of spray breaking over the vessel.

On December 2, an hours' coal consumption trial with three-fifths power, was made on the Clyde, the I.H.P. being 4,653; speed, 13.2 knots; coal per I.H.P. per hour, 2.7 pounds.

Quail.—This 30-knot destroyer, built by Messrs. Laird Bros., Birkenhead, a sister vessel to the *Virago*, was described on page 820 of Vol. VII. The designed displacement is 300 tons, and the I.H.P., 6,000.

She was launched on September 24, 1895, and various preliminary and contractors' trials were made, during which the propellers were modified. In the early part of October, 1896, these trials were finished. On October 27, an official trial was made, but the results were not satisfactory.

On December 11, a satisfactory official trial under full power was made, the speed for three hours being 30.039, and for the six runs over the mile, 30.385 knots. Mean steam pressure, about 216 pounds per square inch; revolutions per minute for three hours, 368, and for the miles, 372.

To determine the coal consumption of this class of destroyers, at the most economical speed, which was decided to be 13 knots, each of them was run for eight hours at this speed. The *Quail* proved herself the most economical, the mean results being as follows. Speed, 13.1 knots; revolutions per minute, 141; I.H.P., 402.4; coal per I.H.P. per hour, 1.6 pounds; coal per day, 6.9 tons.

As these destroyers carry about 80 tons of coal, the radius of action at 13 knots would be about 3,619 nautical miles.

On January 4, according to "Engineering," London, another speed trial was made with slightly modified propellers. Six runs were made over the Skelmorlie mile and the run continued for three hours. The mean speed for the three hours was 30.12 knots, and for the six runs, 30.09 knots. The steam pressure was about 224 pounds per square inch. The I.H.P for the runs over the mile is given as 6,000 by the "United Service Gazette," London.

Thrasher.—A sister vessel of the *Virago* mentioned above,

launched on November 5, 1895, completed her trials on December 15, 1896. A preliminary contractors' trial was made in August, 1896, the speed being about 30.38 knots.

On December 1, an official, full power, coal consumption trial was made on the Clyde. The mean speed for the six runs over the mile was 30.36, and for the three hours, 29.78 knots. The mean steam pressure was 210 pounds per square inch, and the I.H.P. and revolutions per minute, for the three hours, were 6,562 and 359.4, respectively. The coal per I.H.P. per hour was 2.67 pounds. The speed during the three hours was greatly affected by the very strong wind and sea, the vessel at times rolling heavily.

On December 15, another trial was made, the speed for the three hours being 30.015 knots, and the revolutions per minute, 362. The runs over the mile gave a speed of 30 knots with practically 362 revolutions per minute. The steam pressure was 213 pounds per square inch.

A special official trial made on January 13, gave a mean speed of 30.04 knots for the six runs, and 30.34 knots for the three hours, the mean steam pressure being about 208 pounds per square inch.

Sparrowhawk.—The mean speed given on page 842 of Volume VIII, for the three hours' trial on October 16, should be, according to later reports, 30.056 instead of 30.68 knots, the revolutions per minute being 365. The steam pressure was about 214 pounds per square inch. As the speed for three hours is deduced from the revolutions and speed on the mile runs, the speed for the three hours, if the revolutions are correct, should be about 30.29, and not 30.056 knots.

Express.—This 33-knot torpedo boat destroyer, building by Messrs. Laird & Bros., Birkenhead, will, according to latest reports, have twin screw engines which are to develop about 10,000 I.H.P. The displacement will be about 300 tons.

Fame.—This 30-knot torpedo boat destroyer, built by Messrs. J. I. Thornycroft & Co., Chiswick, and launched on April 15, 1896, made a preliminary trial of three hours' duration on October 28, to test the coal consumption, in order to determine the load to be carried on the full power consumption trial, accord-

ing to the new requirements mentioned on page 399 of Volume VIII. The average speed for the three hours was 29.579 knots. A complete description of this type of destroyer was given on page 182 of Volume VIII.

The four hours' full power trial was commenced off Chatham dockyard on January 15, but was interrupted by the breaking of two bolts of the rudder gear.

Mallard.—A sister vessel of the *Fame*, mentioned above, was successfully launched on November 19

Chamois and **Crane.**—These 30-knot destroyers, built by the Palmer's Shipbuilding Co., Howdon-on-Tyne, were launched on November 3 and December 17, 1896, respectively. These destroyers are 215 feet long between perpendiculars; their beam is 20.75 feet; depth, 12 feet 10 inches; mean load draught, 5 feet 5 inches.

Earnest, Locust and **Panther.**—These 30-knot destroyers, built by Messrs. Laird Bros., Birkenhead, were launched on November 7, December 5 and January 21, respectively. The displacement is about 320 tons, and the I.H.P., 6,000.

Otter.—This 30-knot destroyer, built by the Naval Construction and Armaments Co., Barrow, was launched in the latter part of November, 1896. The displacement is about 300 tons, and the I.H.P., 6,000.

FRANCE.

Carnot.—This battle ship, described on page 401 of Volume VIII, has been continuing her trials. Preliminary trials were given on page 625 of Vol. VIII.

On December 5, 1896, a six hours' trial with assisted draft was successfully completed. The mean I.H.P., including the auxiliaries, was 10,203; revolutions per minute, 95; speed, about 18 knots. For a short time, with 98.5 revolutions per minute, the I.H.P. was 11,389. The machinery worked well.

On December 19, the forced draft trial was interrupted by defective boiler feeding apparatus. As the vessel was returning to Toulon, the steering gear became deranged. This was discovered just in time to prevent the vessel from grounding in the entrance to the harbor. She was towed into Toulon harbor by tugs.

On December 29, the full power trial was resumed, three runs being made over the measured mile off the Hyères Islands. With the engines running at 105 revolutions per minute and developing about 16,050 I.H.P., the mean speed was 17.9 knots. The fastest run was made at the rate of 18.5 knots.

The "Yacht," Paris, says that the vessel exceeds the designed displacement by 200 or 300 tons, and the after military mast will be taken out and the number of bridges on the superstructure reduced.

On January 15, a coal consumption trial at 12 knots was made over the measured mile. Only four out of the twelve groups of boilers were used. The mean results for the four runs were as follows. Steam pressure, 142 pounds per square inch; revolutions per minute, 63; I.H.P., 3,109; speed, 11.9 knots. The amount of coal burned is given as 440 pounds per hour per furnace. Taking the grate surface given on page 385 of Volume VII for the eight boilers in use, 361.73 square feet, the coal per square foot of grate would be 19.46 pounds, and the coal per I.H.P., 2.26 pounds per hour.

Further trials will be postponed until one of the under water torpedo tubes has been installed.

A_3.—This battleship will be laid down at Brest, and will be very much like the *Henri IV*, building at Cherbourg. The displacement will be about 8,860 tons, and the speed, under natural draft, 17 knots. Water tube boilers will be used. Probable armament, two 12-inch, six 5.45-inch and four 3.94-inch guns, the last two sizes to be rapid fire guns. Estimated cost, exclusive of armament, $3,689,800.

C_3.—This first class, sheathed, triple screw cruiser will be of the *Jeanne d'Arc* type, and will be laid down alongside of this vessel at Toulon. The length will be 460 feet; beam, 61 feet; and displacement, 11,050 tons.

The proposed armament consists of two 7.48-inch, eight 5.51-inch and twelve 3.94-inch rapid fire guns, twenty-four 3 and 1-pounder rapid fire guns, and two submerged torpedo tubes.

The three engines will be of the vertical, triple expansion type, designed to develop about 28,100 I.H.P. and give the vessel a

speed of 23 knots. Normand water tube boilers will be used. The radius of action, with a bunker capacity of about 2,065 tons, is expected to be 2,000 nautical miles at a speed of 23 knots, and 13,500 miles at 10 knots. The estimated cost is $4,934,754.

D_2 and D_3.—These first class, sheathed, triple screw station cruisers are to be of a new type; one will be built at Lorient and the other, by contract. The Lorient cruiser, D_2, will have a length of 438 feet, a beam of 48 feet, and a displacement of about 5,410 tons. The armament will consist of eight 6.3-inch, and twelve 1.85-inch rapid fire guns; and two submerged torpedo tubes.

The three engines will be of the vertical, triple expansion type, designed to develop about 16,860 I.H.P. and to give the vessel a speed of 23 knots. Normand boilers will be used. The estimated cost is $2,134,962.

Foudre.—This twin screw vessel, built as a torpedo transport or depot ship, and which was described on page 824 of Vol. VII, commenced her trials in September, 1896. There are 24 boilers of the Lagrafel-d'Allest water tube type, arranged in six compartments, with one smoke pipe for two compartments. The triple expansion engines were designed to develop 11,244 I.H.P., and give the vessel a speed of from 18.5 to 19 knots. The following details of the trials are from "LeYacht," Paris, and the "Rivista Marittima," Rome.

On September 30, 1896, a four hours' full power trial was commenced. Everything worked well, the mean speed for about three and a half hours being 19.513 knots. At the end of that time, while the vessel was making the turn at one end of the measured mile, she ran aground. She was successfully floated some time after, and an examination showed that no injury had been done.

On October 28, four runs were made over the measured mile off the Hyères Islands, the mean speed being 19.6 knots. The revolutions per minute were 106, starboard, and 107, port, and the I.H.P., 11,766. The trial was, however, interrupted by the heating of some bearings and by several leaks.

On November 12, a coal consumption trial under forced draft

was made, the average revolutions per minute being 101.5; the I.H.P., 9,883, and the coal per I.H.P., 1.792 pounds. As this consumption was within the contract limits, 1.788 to 2.011 pounds per I.H.P. per hour, there was neither premium nor penalty. The penalty for a consumption below 1.788 pounds was fixed at about $40 for each gramme.

Bruix.—This twin screw armored cruiser, built at Rochefort and launched in 1894, started on her preliminary trials in the middle of 1896. The designed I.H.P. and speed of this vessel were about 8,580 and 19 knots. On one preliminary trial, 7,110 I.H.P. was obtained. Several official trials have been made, but no results published. An attempt at a trial in October, during the heavy gales, showed that this vessel could not use her battery. During the passage from Cherbourg to Brest, the vessel was injured by the high seas. Although the designed speed of 19 knots has never been attained, the vessel is, according to "Le Yacht," Paris, good for 18.5 knots in service, and has been commissioned.

Pascal.—This sheathed, twin screw, second class cruiser, described on pages 189 and 849 of Volume VIII, has completed satisfactory trials. The following particulars are from "Le Yacht," Paris.

The four 6.3-inch guns have been replaced by four 6.49-inch rapid fire guns in sponsons, protected at their bases by 2.13-inch armor. The guns are mounted on central pivot carriages and are protected by 2.13-inch chrome steel shields. A similar protection is given to the ten 3.94-inch guns.

The full power trial was made on December 17, 1896, off Toulon. With 130 revolutions per minute, the engines developed 8,600 I.H.P., and gave the vessel a speed of 19.7 knots. The machinery was not forced, as the contract called for only 8,384 I.H.P. and a speed of 19 knots.

On the trial under natural draft, on December 22, the speed was over 18.5 knots and the I.H.P. below 7,890.

On January 2, a six hours' coal consumption trial was made, running with the starboard engine only, the port one being uncoupled. The trial was intended to be made at a speed of 10

knots, and with 888 I.H.P. The results were as follows. Mean number of revolutions per minute, 76.27; I.H.P., 922; speed, 10.49 knots; coal per I.H.P. per hour, 1.46 pounds.

Tests were made after this trial in coupling and uncoupling the port engine, the former operation taking 13, and the latter, 14 minutes.

The starboard engine was then tried to find the lowest number of revolutions at which the engine would run. This number was found to be 15 per minute, but the slowest speed in regular practice was found to be about 30 revolutions per minute.

On January 11 and 12, a satisfactory 24 hours' trial was made. The mean number of revolutions per minute was 119; I.H.P., 6,085; speed, 17.56 knots; coal per I.H.P. per hour, 1.6 pounds.

K_2.—This third class cruiser will be built by contract, and will be of the *D'Estrées* type, building at Rochefort, but larger and faster. The displacement will be about 2,410 tons. The armament will consist of two 5.45-inch, four 3.94-inch and eight 1.85-inch rapid fire guns. The twin screw engines are designed to develop about 8,380 I.H.P. and to give the vessel a speed of 20.5 knots.

T_1.—This gunboat will be built by contract, and will be similar to the *Surprise*. She will be sheathed, the displacement being about 620 tons; the length, 184.7 feet; beam, 24.27 feet, and the speed, 13 knots. The armament will consist of two 3.94-inch, four 2.56-inch and four 1.46-inch guns. Estimated cost, $264,620.

M_2.—This torpedo boat destroyer will be built by contract. She will be of a new type, but the plans have not yet been completed. The speed is to be 26 knots, and the radius of action at 10 knots, 2,500 nautical miles.

P_{35} and **P_{36}.**—These first class torpedo boats will be of the Normand type, 121.3 feet long, and have a beam of 15.09 feet, a displacement of about 84 tons and a speed of 23.5 knots; twin screws; I.H.P. about 1,480. The armament will consist of two 1.46-inch guns and two torpedo tubes, one of which will be in the bow and the other on a pivot on deck.

Mangini.—This sea-going, twin screw torpedo boat, of which

a description was given on page 148 of Volume VII, completed her trials in November, 1896.

The boat is 147.6 feet long; beam, 14.76 feet; draught, about 7.89 feet; displacement, 127 tons. Contract speed, 25 knots; designed I.H.P., about 2,070. She was built by the Chantiers de la Loire. There are two Du Temple boilers, type 1895. The bunker capacity is about 18 tons of coal, giving a radius of action of 1,800 miles at 10 knots. The complement will consist of 34 officers and men.

On November 6, the full power trial was made off Lorient. A speed of 27.52 knots, or 2.52 over the contract, being realized. This places the *Mangini* next in speed after the *Forban*, the fastest torpedo boat in the French navy.

GERMANY.

Kaiser Friedrich III.—It may be noted that this battleship, see page 827, Vol. VII, and table facing page 556, Vol. VIII, will be the first vessel to have the new Krupp rapid fire guns of 9.45-inch caliber.

Odin.—This fourth class battleship, the eighth of the *Siegfried* class, completed her trials in November (?), 1896. This vessel was launched at Danzig on November 5, 1894, and made her trials during 1896. No data of the trials are available.

The following description is taken from "Mittheilungen aus dem Gebiete des Seewesens," Pola. In general, the *Odin* is a sister vessel of the *Aegir*, described on page 604 of Volume VII. The length between perpendiculars is 236.2 feet; beam, 50.53 feet; mean draught, full load, 17.38 feet; corresponding displacement, 3,474 tons. The earlier vessels displaced 3,440 tons. In the *Aegir* and *Odin*, the side armor extends only around part of the vessel, a central citadel and protective deck, all of Krupp's nickel steel, completing the protection.

The armament consists of three 9.45-inch, 45 caliber guns; ten 3.46-inch rapid fire guns; six machine guns, and three 17.72-inch torpedo tubes, one in the bow under water and the others in broadside above water.

The twin screw, triple expansion engines are the same as those

of the *Aegir*, with cylinders 26, 40 and 63 inches in diameter, the stroke being 29.5 inches. The designed I.H.P. is 4,734, and the expected speed is given as 17 knots. There are eight water tube boilers, fitted to burn *masut*, or liquid fuel refuse.

The coal capacity of the *Odin* has been increased to about 260 tons, that of the older vessels being only about 220 tons. The complement consists of 166 officers and men.

Hela.—This twin screw despatch vessel, the largest one in the German Navy, has completed her official acceptance trials. The following description of the vessel and data of the trials are from the "Marine Rundschau," Berlin.

The *Hela* was laid down at the Weser Co.'s yard, Bremen, on December 5, 1893, and was known as ship *H*. She was successfully launched on March 28, 1895, and, after finishing the contractors' trials, was preliminarily accepted on January 18, 1896, and taken to the dockyard at Wilhelmshaven.

The hull is built of steel, with a protective deck and coffer dams. The length between perpendiculars is 328.1 feet; beam, 36.09 feet; mean draught, 15.42 feet; displacement, about 1,970 tons. The designed I.H.P. of both engines was 5,918, and the speed, 20 knots. The ram is very long. There are two pole masts and one smoke pipe. The armament consists of four 3.46-inch and six 1.97-inch rapid fire guns; and three 17.72 inch torpedo tubes, one submerged in the bow and two above water in broadside.

There are six boilers. The diameter of the propellers was fixed at 10.66 feet as a result of preliminary trials with the original and larger propellers. The pitch is 13.45 feet. The machinery is very light, every effort having been made to reduce the weight. The coal bunker capacity is about 350 tons.

The complement consists of 168 officers and men.

On May 3, 1896, the vessel was put in commission for the official trials. On May 16, the vessel was to steam to Kiel, by way of Cape Skagen, Denmark, but, owing to the breakdown of the starboard engine, the vessel went to Kiel through the Emperor William Canal with the port engine. Repairs were finished on May 30, when trials were made over the measured mile to determine the most advantageous pitch for the propellers. These

trials showed that the propellers were too large, causing excessive vibration. The diameter and pitch were then fixed as given above, the vibration being reduced to very little.

During the contract trials, on July 18, the H. P. cylinders of both engines were damaged, and caused a postponement of the trials until August 3. The trials were finished on September 11, 1896.

The following are the results of the various runs over the measured mile, the first two trials having been made under forced draft. The radius of action is based on a bunker capacity of 344 tons of coal.

Revolutions.	I.H.P.	Speed, knots.	Radius, N. M.
176.5	5,780	20.428	. . .
166.9	4,746	19.528	1,584
158.8	4,062	18.504	1,762
141.75	3,855	16.549	2,267
133.25	2,347	15.474	2,488
94.85	975.4	11.170	3,503

The coal consumption, for speeds varying between 15.474 and 18.504 knots, and for 11.17 knots, was obtained from 24-hour trials. For the former, which is called "quick speed for long distances," the coal per I.H.P. per hour was found to be 1.9 pounds, and for the latter, 11.17 knots, which may be called "service speed," the coal consumption was 2.18 pounds per I.H.P. per hour.

With one engine stopped, and using all the steam supplied by the six boilers under an air pressure of .47 inch, the maximum performance of the other engine was 2,604 I.H.P. with 154.3 revolutions per minute, the mean speed of the vessel being 15.7 knots.

When steaming, the stern is raised. The bow and stern waves are not considerable. The height of the wave, when steaming at the maximum speed of 20.4 knots, was 3.94 feet. The vessel has great stability, as was shown during full speed with the rudder hard over, the list then not exceeding 3 degrees.

When steaming with one engine, the rudder must be put over 10 degrees to steer a straight course. The diameters of the

turning circles, at a speed of 11 knots, with the rudder hard over, were found to be 2,080 feet to starboard and 2,067 feet to port. At 16.5 knots, the diameters were 2,067 to starboard and 2,132 feet to port. When steaming ahead with one engine and backing at the same speed with the other, these diameters were reduced, on an average, from 328 to 656 feet, respectively. The vessel's headway was stopped in the following times and distances, the time being counted from the backing of the engines with the full available power.

Speed.	Elapsed time.	Distance.
5 knots.	32 seconds.	190 feet.
7 "	45 "	312 "
9 "	50 "	328 "
11 "	53 "	361 "
19 "	75 "	984 "

HOLLAND.

Friesland and **Holland.**—These protected cruisers, described on page 829 of Volume VII, were successfully launched on November 5 and October 4, 1896, respectively. The *Friesland* was launched by the Netherlands Steamship Co. at Feijenoord and the *Holland*, at the dockyard at Amsterdam.

JAPAN.

Armored Cruisers.—Contracts for two armored cruisers were let on December 29, 1896, one to the Wm. Cramp & Sons Ship and Engine Building Co., Philadelphia, and the other to the Union Iron Works, San Francisco.

Both vessels will have twin screws and be built of steel. The armament, to be supplied by Japan, will consist of two 8-inch guns; ten 4.7-inch, twelve 12-pounder and two 2½-pounder rapid fire guns; and five torpedo tubes, one of which will be in the bow, under water, and the other in broadside. The engines will be of the vertical, triple expansion, four cylinder type. Each vessel will have twelve single-ended cylindrical boilers in three compartments, and one smaller auxiliary boiler above the protective deck.

The working pressure will be 155 pounds per square inch. The designed speed is 22.5 knots.

The principal data so far fixed for the cruiser to be built at Cramp's shipyard are as follows. Length between perpendiculars, 374.5 feet; beam, 48 feet 9 inches; mean draught, 17 feet 9 inches; displacement, 4,900 tons; I.H.P., 17,000; grate surface, 920, and heating surface, 26,000 square feet.

The Union Iron Works cruiser will be a little longer, but draw slightly less water. The principal data are as follows. Length over all, 405 feet; length between perpendiculars, 376 feet; beam, 49 feet; mean draught, 17 feet 7.25 inches; displacement, 4,760 tons; I.H.P., 15,500; grate surface, 792, and heating surface, 22,000 square feet.

Torpedo Boat Destroyers.—Messrs. Yarrow & Co. have received an order to build two destroyers, the guaranteed speed being 31 knots.

Torpedo Boats.—Mr. Normand, Havre, is to build five seagoing torpedo boats.

NORWAY.

Harald Haarfagre.—This coast defense vessel, one of the two described on page 635 of Volume VIII, was built by Sir W. G. Armstrong & Co., and successfully launched on January 4.

The dimensions of the vessel as finished and the armament are given by "Engineering," London, as follows. Length, 280 feet; beam, 48.5 feet; draught, 16.5 feet; displacement, 3,500 tons. The armament will consist of two 8-inch, four 4.7-inch (?), six 12-pounder and six 1½-pounder guns, and two 18-inch torpedo tubes.

The belt armor will be from 4 to 7 inches thick, and the conning tower, 6 inches thick.

PORTUGAL.

1,770-ton Cruisers.—The contract for these two protected, sheathed cruisers, mentioned on page 852 of Volume VIII, has been given to the Forges et Chantiers de la Mediterranée, Havre, according to "Le Yacht," Paris, from which the following description is taken.

Length between perpendiculars, 246 feet; beam, 35.43 feet; depth, 23 feet; draught, 14.27 feet; displacement, 1,771 tons; I.H.P., about 2,615; speed, natural draft, 15 knots.

The protective deck will be .79 inch thick on the inclined sides. A double cofferdam will reach from the protective deck to 3.28 feet above the water line and will extend from the torpedo workshop forward to the stern. The hull will be sheathed with teak and coppered. The stem and stern post will be of bronze. There will be two low masts rigged as a barkentine.

The armament will consist of two 5.9-inch, four 4.7-inch and eight 1.85-inch Canet rapid fire guns, two machine guns and one torpedo tube forward.

The twin screw engines will be of the vertical, triple expansion type, and be placed in separate compartments. The boilers will be of the Normand & Sigaudy type, placed in two compartments. There will also be an auxiliary boiler.

The electric light and power plant will consist of two dynamos of 200 amperes and 80 volts. The ammunition hoists will be electric.

RUSSIA.

New Battleship and **Cruiser.**—A battleship, of about the same type as the *Rostilav*, described on page 853 of Volume VIII, is to be built at Nicolaieff. The displacement is to be 8,660 tons, the I.H.P., 8,384, and the speed, 16 knots.

A cruiser of 13,779 tons displacement, which will be an enlarged *Rossija*, described on page 637 of Volume VIII, has been laid down at the Baltic Works, St. Petersburg.

Sissoi Veliki.—This twin screw battleship, built at the new Admiralty Works, St. Petersburg, and launched on June 2, 1894, had her trials on October 17, 1896.

The length on load water line is 345 feet; length between perpendiculars, 332 feet; beam, 68 feet 10 inches; draught, 22 feet; displacement, about 8,880 tons. The hull is built of steel. There is a complete belt at the water line, 15.75 inches thick amidships and tapering to 11.8 inches at the ends, with athwartship bulkheads 11.8 inches thick; above this is a central casemate, 5 inches thick. The turret armor is about 11 inches thick. The protec-

tive deck, 3 inches thick, is curved, the crown being about 3 feet above, and the edges at the sides about 5 feet below the water line.

The armament consists of four 12-inch guns, arranged in pairs in the two turrets; six 5.9-inch, twelve 1.85-inch and four 1.46-inch rapid fire guns; two machine guns; and six torpedo tubes, one forward, one aft, and four in broadside. The arc of five of the turret guns is 230 degrees, and that of the 5.9-inch guns 120 degrees.

The twin screw triple expansion engines were designed to develop 8,500 I.H.P. and give the vessel a speed of 16 knots.

There are two smoke pipes and two military masts. The complement comprises 32 officers and 550 men. The coal supply is about 800 tons. The cost of the vessel is about $4,000,000.

The following data of the trial are from the "Marine Rundschau," Berlin.

The draught was 23.75 feet, or 1.75 feet more than the contract draught. The corresponding displacement was 9,608 tons. The mean speed, obtained from runs over the measured mile, was 15.65 knots. The revolutions per minute varied from 82 to 89, and the steam pressure in boilers, from 100 to 125 pounds per square inch. The average results obtained from eight sets of indicator cards taken from each engine were as follows. Starboard engine, steam pressure, 110 pounds; revolutions, 87.5; I.H.P., H.P. cylinder, 1,357, I.P., 1,224, L.P., 1,625; I.H.P., collective, 4,206; port engine, steam pressure, 111 pounds; revolutions, 87.6; I.H.P., H.P. cylinder, 1,253, I.P., 1,291, L.P., 1,745; I.H.P., collective, 4,289.

The total I.H.P. was 8,495, or 5 below the contract requirement. The average I.H.P., computed from four sets of indicator cards, taken while running over the mile, was, however, 8,635. For this power the speed was, as stated above, 15.65 knots.

Admiral Senjavin.—This twin screw coast defense vessel, a sister to the *Admiral Oushakoff*, described on pages 175 and 839 of Vol. VII, completed her trials on October 30, 1896.

The vessel was built at St. Petersburg and launched in 1894. The length over all is 278 feet; beam, 52 feet; draught, mean, 17

feet; displacement, 4,126 tons. There is a complete belt armor, 10 inches thick; above this, there is a casemate, with from 7 to 8 inches of compound armor. The protective deck varies from 2.5 to 3 inches.

The armament consists of two large guns, one forward and the other aft, the caliber of which is given as 9 and 10 inches; four 5.9-inch and six 1.85-inch rapid fire guns; eight machine guns; and four torpedo tubes.

The twin screw, triple expansion engines were designed to develop 5,000 I.H.P. and give the vessel a speed of 16 knots. They were built by Messrs. Humphreys, Tennant & Co., Deptford.

The coal supply is about 400 tons, and the complement, 318 officers and men.

The following data of the twelve hours' trial on October 30, 1896, are from the "Marine Rundschau," Berlin.

The average results from eleven sets of indicator cards taken from both engines gave 5,254 I.H.P., or 254 over the contract requirement. The steam pressure was 123.5 pounds per square inch, and the revolutions per minute, starboard, 130.87, port, 130.4. The above I.H.P. was distributed as follows.

Starboard engine, H.P. cylinder, 835, I.P., 875, L.P., 928; total, 2,638. Port engine, H.P. cylinder, 815, I.P., 899, L.P., 902; total, 2,616.

The I.H.P. of the sister vessel, the *Oushakoff*, on the twelve hours' trial was 5,764.8, and the speed, 15 knots.

Rossija.—This triple screw armored cruiser, which was described on page 637 of Volume VIII, ran aground off Cronstadt in November, 1896, and is now frozen in. The following account is from the "Times," London.

"Our St. Petersburg correspondent telegraphed last night, November 9: 'The new Russian cruiser the *Russia*, which was only launched in the Neva on May 12, is still on a sand bank off Cronstadt. Large dredgers have been ordered from Helsingfors to dig a passage and try to get her afloat. The *Russia* is the largest ship in the Russian Navy, and was built at the Baltic works. No other Russian war vessel has ever been ready for sea so soon after leaving the slips. This is due to the anx-

iety of the Admiralty to get her abroad this year before the ice closes up the river. After a short stay at Cronstadt the cruiser was to go to Libau to complete her equipment for the Mediterranean. For two weeks high water was awaited in the Neva to permit her to steam down to Cronstadt. On Saturday, November 7, a strong west wind and storm, which flooded the lower parts of St. Petersburg, gave the desired opportunity. Twelve tugs towed her into the sea canal, and the cruiser then took her own course, the weather being too rough for the tugs to accompany her all the way. According to the account published in to-day's 'Petersburger Zeitung,' a piston of an air pump broke, and something occurred to disable one of the three engines. In spite of this, the cruiser reached Cronstadt Roads and let go two of her anchors, but the wind changed to the north and, in turning, the vessel grounded near one of the mid-water forts. Unfortunately, the wind dropped suddenly, the water ebbed away, and the cruiser, in spite of every effort, sank still deeper into the sand. It is a question whether she can be got off before the ice prevents navigation."

The "Yacht," Paris, says that after all efforts had failed to get the cruiser off, preparations were made to protect her from the ice. Piles were driven in the sand all around the vessel, and a wall of beton blocks was seriously considered. The cruiser is resting easily and safely, as far as known, and, it is to be hoped, will get off safely in the spring.

Svetland.—This twin screw protected cruiser, built by the Forges et Chantiers de la Mediterranée at their Graville yard, near Havre, France, was successfully launched on Sunday, December 6, 1896. The vessel was laid down on December 1, and the first rivet driven December 7, 1895.

The following details are from "Le Yacht," Paris:

The length is 331.4 feet; beam, 42.65 feet; draught, 18.76 feet; displacement, 3,767.5 tons. The protective deck varies from .82 to 1.48 inches in thickness, a splinter deck being below this over the machinery spaces.

The armament will consist of six 5.9 Canet and ten 1.85-inch Hotchkiss rapid fire guns, and four torpedo tubes.

The vertical triple expansion engines are designed to develop 8,384 I.H.P. under forced draft and give the vessel a speed of 20 knots. There are to be eighteen Belleville boilers. There will be three smoke pipes and two light masts. The normal coal supply will be about 980 tons.

Diana and **Pallada.**—These cruisers, building at the Franco-Russian dockyard, St. Petersburg, will have three screws and a speed of 20 knots. The length will be 414 feet; beam, 55 feet; draught, 21 feet; displacement, about 6,630 tons. They are to be launched not later than November, 1897.

The engines will be of the vertical triple expansion type and are designed to develop about 11,600 I.H.P. The boilers will be of the Belleville type.

With the exception of the boilers and their fittings, which will be made abroad, everything is to be of Russian make.

Tri Sviatitelia (Three Saints).—This twin screw battleship, belonging to the Black Sea Fleet, and described on page 211 of Volume VI, completed her official trials in November, 1896. She was launched at Nicolaieff in 1893.

The length is 357.5 feet; beam, 72.2 feet; draught, 27.5 feet; displacement, 12,480 tons. The armament consists of four 12-inch guns, in pairs in two turrets; eight 5.9-inch and four 4.7-inch rapid fire guns in the casemate; fifty-six smaller rapid fire and machine guns, and six torpedo tubes, two of which are under water. There are two military masts.

The machinery was built by Messrs. Humphreys & Tennant, Deptford, the triple expansion engines having cylinders 43, 62 and 96 inches in diameter, with a stroke of 51 inches. The total cooling surface of the two condensers is 18,500 square feet.

There are fourteen single-ended, three furnace boilers, the total grate surface being 1,050 and heating surface, 31,024 square feet; and two auxiliary boilers with a total grate surface of 44 and a heating surface of 1,084 square feet. The boilers are arranged in four compartments, with two smoke pipes for all boilers. The tubes are 3 inches, and the furnaces, 50 inches in diameter. The propellers are 17 feet in diameter, and the pitch can be varied from 22 to 25.5 feet.

The engines were designed to develop 8,500 I.H.P. under natural draft, and give the vessel a speed of 16 knots.

The twelve hours' trial in November, 1896, was made off Sebastopol, and was satisfactory. The results, as published, were 9,500 I.H.P. and a speed of 18 knots, both contract power and speed being exceeded.

Standard.—This imperial yacht, described on page 434 of Vol. VII, made her final trial trip in the Kattegat on October 31, 1896.

The trial lasted twelve hours, and the machinery worked well, without causing any vibrations of the vessel. The mean speed was 21.18 knots, obtained by observation of landmarks. This speed is 1.18 knots above the contract, and was attained with about 92 revolutions per minute. The maximum speed was fully 22 knots.

SPAIN.

Furor and **Terror.**—These torpedo boat destroyers, the launching of which was mentioned on page 856 of Volume VIII, have successfully completed their official trials. They were both built by Messrs. J. & G. Thomson, Clydebank.

The length is 220 feet; beam, 22 feet; depth, 13 feet; load draught, 5.5 feet; corresponding displacement, 380 tons; contract speed, forced draft, with a load of 75 tons, 28 knots. This speed to be maintained for four runs over the measured mile and for two hours after. There are eleven water tight compartments. The bunker capacity is 80 tons, giving a steaming radius of 2,000 nautical miles at 13 knots.

The following additional particulars are from "Engineering," London.

These vessels are fitted with various appliances to render existence on board more endurable in a tropical climate. There is a sheathing of teak planks over the steel upper deck, complete awnings all forward and aft, and an installation of electric ventilating fans throughout the crew's and officers' quarters. The ships are lighted by electricity throughout. The armament consists of two 14-pounder quick firing Maxim-Nordenfeldt guns, the one mounted forward and the other aft; two 6-pounders of

the same pattern on the broadsides; and two automatic Maxim guns on the port and starboard bow. There are also two 14-inch deck-pivoted torpedo tubes on the Schwartzkopff system.

The twin screw engines, in one compartment, are of the vertical, four cylinder, triple expansion type, designed to develop about 6,000 I.H.P. The cylinders, which are independent castings, are bolted together, and are stayed transversely between each other, and also to gusset plates in the wings, connecting the deck beams and the frames. A pair of struts extends from each H.P. cylinder to gusset plates between the deck and forward bulkhead of the engine room, and similar ties secure each of the after L.P. cylinders to the hull. The cylinder bottoms are separate castings, and they, together with the cylinder covers and the top and bottom covers of the valve casings, are of cast steel. The pistons are of forged steel, turned all over, and are fitted with two Perkins' metal rings. The piston and connecting rods are of wrought steel and are hollow. The crosshead pin is fixed into the forks of the connecting rod, and is of steel, case-hardened. The piston rod guide and head are forged with the rod, and the guide has machined recesses on the ahead side, which are filled with white metal. The ahead guide faces are of cast iron, with water circulation at the back, and the astern faces, of bronze.

Each bedplate consists of two angle-bar-shaped steel castings, which extend the whole length of the engine and rest on longitudinal frames extending between the engine room bulkheads. The main bearing frames extend between, and are checked into these castings, being secured to them, as well as to the floors of the vessel, by fitted bolts. There are eight main bearings; wrought steel columns, braced together near the center, form the connection between the cylinders and these frames.

The crankshafts and pins are hollow and of steel. The H.P. and I.P. cranks are opposite each other, and each pair of cranks is forged in one piece. Balance weights are fitted on the crank webs. The thrust and propeller shafts are hollow and of steel. The propellers are solid, three-bladed, and of manganese bronze.

The condensers are of brass and are placed in the wings.

Water flows through the tubes by natural means when the vessel is running. Two centrifugal circulating pumps are fitted for use when starting or manœuvring. The main air pumps are single-acting and are worked by cranks on the forward ends of the crank shafts.

The working platform is at the forward end of each engine. There are four boilers of an improved Normand type, placed in two water tight compartments. These are fired from two stokeholds, the furnaces of the forward and aft boilers discharging into separate funnels, while those of the two middle boilers lead up to a common central funnel of larger size. The working pressure in boilers is 300 pounds, and at the engines, 250 pounds per square inch. The tubes are of steel and galvanized. There are two single-breasted blowers on the bunker bulkheads in each fire room. The closed fire room system of forced draft is used.

The boiler feed arrangement consists of a main and auxiliary system of feed pumps and pipes, which are separate and independent of each other. The main system consists of two Weir's special feed pumps, placed at the forward end of the engine room. Each draws from the hotwell tank through a Harris' patent feed water filter, discharging either to the boilers in one compartment direct, or through one of two Weir's feed water heaters, situated on the forward engine room bulkhead between the pumps. The feed water discharge to each boiler is controlled by a special automatic feed regulating arrangement. For the auxiliary system, a similar feed pump is placed in each boiler room, and each pump is connected by separate pipes with the reserve fresh water tank, the hotwell tank and the sea, and discharges direct to the boilers. The pump in the after boiler room also discharges to the deck, and serves the purpose of a fire pump.

The main steam pipes are of steel galvanized. An independent pipe extends between each boiler room and one set of engines, and each pipe is fitted with a stop valve and steam separator on the engine room bulkhead. There is also a connecting pipe between the main steam pipes, and an equilibrium valve at each H.P. cylinder. An auxiliary steam pipe supplies all auxiliary

engines, except the main feed pumps, the exhaust being led to either of the main condensers. An auxiliary air pump is fitted to each circulating pump, and is connected to the bottom of the main condensers, discharging into the hotwell tanks.

There is an evaporator with a capacity of 3,150 gallons of fresh water per day, and a distiller with a capacity of 560 gallons of potable water per day.

Besides the forced draft, full power trials, these vessels were required to make another two hours' trial, during which a speed of 21.5 knots was to be maintained with an air pressure not exceeding $\frac{3}{4}$ inch.

After some preliminary trials, the *Furor* successfully passed her official forced draft trial on October 30, 1896. The mean speed for four runs over the measured mile was 28.25 knots, with the contract load of 75 tons on board. On November 21, running with $\frac{3}{4}$-inch air pressure, the speed was 21.65 knots.

The *Terror* completed her forced draft trials on November 20, having previously obtained about 22.5 knots on the natural draft trial. The contract speed of 28 knots was exceeded without difficulty. In the teeth of a strong southwesterly breeze the vessel behaved admirably, and there was almost a total absence of vibration.

Audaz, Osado, ———, and ———.—These four 30-knot torpedo boat destroyers are building by Messrs. J. & G. Thomson, and will be similar to but larger and faster than the *Furor*. The "Revista General de Marina," Madrid, gives the following particulars of these destroyers.

Length, 225 feet; beam, 22.5 feet; depth, 13.5 feet; load displacement, about 415 tons; speed, 30 knots; and I.H.P., about 7,500. The bunker capacity will be about 95 tons, and the cost about $463,000 each.

Torpedo Boat Destroyers.—According to "Engineering," London, and other papers, Messrs. J. and G. Thomson, Clydebank, have received an order from the Spanish Government to build sixteen torpedo boat destroyers similar to the *Furor* and *Terror* described above.

SWEDEN.

Njord and **Thor.**—The hull of the first of these coast defense vessels of the *Odin* type is building by the Lindholmen's Fabrik, Göteborg, and the machinery by Motala, and the hull and machinery of the second, by the Bergsunds Maschinenfabrik, Stockholm. These vessels will have a displacement of about 3,400 tons, and the I.H.P. will be about 3,700.

The armament of each vessel, according to the "Mittheilungen aus dem Gebiete des Seewesens," Pola, will consist of two large guns, six 5.9-inch and ten 2.24-inch rapid fire guns, four machine guns and one under water torpedo tube. The large guns of the *Thor* will be 9.84-inch Canet guns; those of the *Njord* will be built at Bofors, Sweden.

MERCHANT STEAMERS.

Grand Duchess.—The following additional particulars of this Plant Line steamer, described on page 205 of Volume VIII, and of her trials, are compiled from the "Marine Journal" and "Seaboard," New York.

The registered gross tonnage is 5,017.9 tons. Carrying out an idea of Mr. H. B. Plant, the president of the Plant System of Railroads and Steamships, the greatest beam is 25 feet forward of the middle of the length of the vessel. All hull frames and side plates are carried up to the promenade deck. While this increases the cost, it adds 8.5 feet to the freeboard and 20 per cent. to the longitudinal strength. On account of the great cost of this plan, it has not heretofore been adopted in coastwise steamers.

There are nine water tight bulkheads, running up to the main deck. As an additional factor of strength, the usual shaft tunnels have been improved upon; they run from the double bottom to the next steel deck above, on both sides, like the truss of a bridge. This not only strengthens the ship aft but lessens the vibration. The pilot house deck—the deck above the plating of the ship—is cut fore and aft of the engines, so as to provide for the expansion and natural spring of the deck.

The interior fittings are of the most approved patterns, and the general finish, very fine. The upper berths in all state rooms turn up, as in Pullman sleeping cars, and are made of solid mahogany.

The fire equipment has thirty hose connections, with hose attached to each fire plug. There is a pressure of 50 pounds per square inch on the pipe system, so that in one minute a stream can be turned on from any one plug. The hose is laid in a fore-and-aft position so that one man can manage it, besides open the valve and get a stream of water at once. The law requires this ship to have 600 feet of hose, but 1,500 feet have been provided.

A submerged military rudder of cast steel frame with a single steel plate, and the first built for a merchant ship of the United States, is fitted, in view of her conversion at any time into an auxiliary cruiser. On each end of the rudder quadrant there are 12.5 inches of solid rubber to take the shock off the rudder.

All houses on the promenade and pilot house decks are of steel, with steel doors, each of which cushions on three seats when shut, and steel shutters for each window, the first to be applied to a coastwise ship.

The *Grand Duchess* is heated throughout by the vacuum system, and is the first vessel to adopt this system. All pipes are of brass. There is an exhaust system of ventilation, put in by the Sturtevant Co., Boston, Mass. By this system the air can be taken out of any room or space on the main deck, and also out of the cargo spaces, every two and one-half minutes. The exhaust pipes run through each room, the dining saloons, the living quarters for the crew, and also through the engine and fire rooms. This system in the holds is a great advantage in carrying perishable cargo.

The twin screw, quadruple expansion engines have cylinders 24, 34, 48 and 68 inches in diameter, the stroke being 42 inches. The usual working pressure at the engines is about 225 pounds per square inch. The pistons and all the working parts of the engines are of open hearth steel. The crank and propeller shafts are of steel, hollow, and were made by the Bethlehem Iron Co. The propellers are of manganese bronze and work inwards.

The main steam pipes are of rolled steel with a slip joint designed and patented by Mr. Sommers N. Smith, the Superintendent of the Newport News Co. In this joint, the center is made fast and the pipe from both the boiler and engine slides on that, thus bringing no strain on cylinder or boiler. The flanges of all steam and feed pipes are recessed one-quarter of an inch and then bolted together on a smooth face.

There are four endless belt conveyers to trim the coal across the ship in the bunkers. The coal is hoisted from alongside by a double drum hoister, located on each side of the ship; these

hoisters being operated by an 8-H.P. electric engine, and geared with friction clutch.

A Baird evaporator and See's ash ejectors are fitted. All engine rods are fitted with Katzenstein's metallic packing. The Babcock & Wilcox boilers were described and illustrated on page 702 of Volume VII. The total grate surface of the eight boilers is 460 and the heating surface, 20,300 square feet. The total weight of the boilers, including fittings, pipes and water, is about 263.4 tons. The boilers and bunkers occupy 54 feet of the length of the vessel. There is an auxiliary boiler of the Babcock & Wilcox type, with grate surface of 36, and a heating surface of 1,000 square feet. All of these boilers have been built for a working pressure of 250 pounds, but at present only 225 pounds will be used. All boilers and steam pipes are covered with Magnesia Sectional Covering.

The draft can be forced to the extent of one inch water pressure by means of two Sturtevant blowers. These discharge the air into ducts running fore and aft in the center of the ship and behind the boilers, branch ducts from each supplying air to the under side of each grate. That this air may be equally distributed, it enters both sides of the ash pit, blast gates being provided for its regulation.

The requirements for the trial trip for acceptance, were to make 17 knots an hour in average sea weather with 1,700 tons of cargo aboard and 500 tons of coal in bunkers. This speed was exceeded by one-half knot. On the first trip from Newport News to New York, in November, in the teeth of a 30-mile gale, the *Grand Duchess* steamed steadily at 16.5 knots with ease.

On the builders' trial, the average speed maintained for four hours was 17.9 knots, the engines making 123 revolutions per minute. The draught of the vessel was 16 feet 8 inches forward and 18 feet 5 inches aft, there being 1,000 tons of water ballast and 1,250 tons of coal on board. Over the same course on the same day, with 115 revolutions per minute, the average speed for four hours was 16.12 knots.

For the earlier service trips, the limit of revolutions was fixed at 105 per minute, giving a speed of about 14.5 knots. On the

trip up from Savannah and ending in New York on January 2, the revolutions were increased to 113 per minute. The distance between light ships is about 652 nautical miles, which was made at the rate of 15.34 knots. On the last northward trip ending on January 30, the run from dock to dock was made at the rate of 16.2 knots, no allowance being made for loss while steaming at half speed for a great part of the distance from Sandy Hook to the dock at New York.

Creole.—This passenger and freight steamer of the Cromwell Line, described on page 857 of Volume VIII, has been put into service.

On the trip from the yard of the builders, the Newport News Shipbuilding and Dry Dock Co., to New York, the contract speed of 14.5 knots was made with 71 revolutions per minute.

Arcadia and **Arabia.**—These single screw steamers, built by Messrs. Harland & Wolff, Belfast, Ireland, for the Hamburg-American Steam Packet Co., have been launched. The *Arabia* was launched on December 21, 1896, and the *Arcadia*, sometime previously. They have a gross tonnage of about 5,450. The engines are of the quadruple expansion type.

Canada.—This twin screw steamer was built by Messrs. Harland & Wolff, Belfast, Ireland, for the Dominion Line for the mail service between Liverpool and Halifax, in connection with the Canadian Pacific Railroad.

The length is 500 feet; beam, 58 feet; depth, 31 feet; gross registered tonnage, 8,806 tons; average speed to be maintained on the round trip, 20 knots. The peculiar under water stern construction, adopted by the builders for many of their twin screw steamers, is used in the *Canada*. Instead of the usual continuous keel, with propellers on either side, the bottom rises from some distance forward of the propellers to the rudder post, with a circular opening cut out for the propellers. By this arrangement the propellers are brought nearer to the center line of the vessel, the port propeller being always forward of the starboard one. The base of the rudder post is several feet above the plane of the bottom extended.

The twin screw engines are of the vertical triple expansion

type, with cylinders 28.5, 47.5 and 77 inches in diameter, the stroke being 54 inches. The designed I.H.P. is 7,550. There are seven cylindrical boilers.

A satisfactory trial trip was made on September 26, and the vessel started on her first voyage on October 1, 1896.

Crescent City.—This steel freight steamer, building by the Chicago Shipbuilding Co., S. Chicago, Ills., for Mr. A. B. Wolvin and others of Duluth, Minn., was launched on February 2. The hull will be like the *Zenith City*, although longer, built by the same company and described in detail on page 765 of Volume VII.

The length over all will be 426 feet, on keel, 406 feet; beam, 48 feet; and depth, 28 feet.

A quadruple expansion engine, built by the same company, will be fitted, the first large lake steamer to be fitted with this type. The cylinders will be 19, 28, 41 and 60 inches in diameter, the stroke being 42 inches. Babcock & Wilcox water tube boilers will be installed, the working pressure to be 250 pounds per square inch.

This steamer is to be completed about July 15. The cost is given as $250,000.

Empire City.—This freight steamer, building by the Cleveland Ship Building Co., Cleveland, Ohio, for Mr. A. B. Wolvin, will be generally like the *Crescent City* described above. No particulars are given, except that a quadruple expansion engine and Babcock and Wilcox boilers will be installed.

Ohio.—This steamer of the Wilson Line, fitted with Belleville boilers, has been very prominent in the various discussions on "Water Tube Boilers." Some data of the engines and boilers were given on page 619 of Volume VII. The following additional information is from "Engineering," London, and from a letter sent to that paper by Mr. A. E. Seaton of Earle's Shipbuilding and Engineering Co., Hull.

The four Belleville boilers are arranged with athwartship fire-rooms. There are eight tube sections in each boiler, the length of each section being 7 feet 6.5 inches. There are twenty 4.5-inch tubes in each section, the tubes being arranged in pairs. The

total grate surface is 192, and the heating surface, 6,013 square feet. The working pressure in boilers is 250, and at the engines, 200 pounds per square inch. The I.H.P. of the quadruple expansion engines is given as 1,500.

Mr. Seaton, in reply to a statement that the machinery was not specially arranged for the Belleville boilers, says: "The engines of the *Ohio*, which are of the four crank quadruple type, were made specially for a working pressure of 200 pounds per square inch and to work with the Belleville boilers. As a matter of fact, the engines appear to have been admirably suited for the Belleville boilers in question, for from the day they were started up to the present time they have worked without a hitch, and, fortunately for the engineers, they can be left to take care of themselves, as has not unfrequently been done, owing to the engineers' services being required elsewhere. From a set of diagrams, taken at sea under ordinary conditions, the water consumption is shown to be 12.3 pounds per I.H.P. per hour. I regret to say, however, that the consumption of fuel is not so satisfactory, perhaps in consequence of its being coal from the South Yorkshire collieries, for the efficient burning of which I do not consider the Belleville arrangement suitable."

Cambrian.—This steel steamer, built by Messrs. Wm. Gray & Co., West Hartlepool, for the Wilson-Furness-Leyland Line, to trade between London and New York, was launched on October 9, 1896. The following description is from "The Marine Engineer," London.

Length over all, 464 feet; beam, 49 feet; depth, 34 feet; measured capacity, 10,000 tons. The vessel has seven water tight bulkheads, and the double bottom extends the whole length of the vessel. Including the after peak, the double bottom has a capacity of over 1,200 tons of water ballast. There are four masts fitted with fore-and-aft sails. There are seven cargo hatches and nine steam winches, with extra whipping drums, for the handling of cargo. Movable stalls are fitted for about 700 head of cattle.

The single screw, triple expansion engine, built by the Central Marine Engine Works, is the largest built in the Hartlepools.

The designed I.H.P. is about 4,000, and the expected speed of the vessel, 14 knots. The cylinders are 32, 54 and 86 inches in diameter, the stroke being 54 inches. As the vessel is to be continuously engaged in the North Atlantic service, all parts of the engines, especially the bed plates, engine frames and condensers, have been made very heavy. The bed plate is of box section and weighs 31 tons. The condenser runs the whole length of the engine, and is bolted to the bed plate. There is a horizontal joint at about the middle of its height. The cylinders are supported in front by two large cast iron columns of box section with a wrought iron column between them. The pistons are made exceptionally deep and are fitted with Mudd's packing rings. The connecting rods are 10.5 feet long between centers and have large bearings. The air pump is worked from the main engine. The centrifugal circulating pump is driven by an engine on each side of it. There are two Weir feed pumps, a Mudd evaporator, and other auxiliaries.

The body of the crank shaft is of ingot steel, and the whole weighs 20 tons. There are six large main bearings lined with white metal. The thrust shaft is also of ingot steel, with an unusually large collar surface. The propeller has four bronze blades.

There are three cylindrical boilers, the working pressure being 185 pounds per square inch. Howden's forced draft system is used.

————.—This passenger and freight steamer is building by the Delaware River Iron Ship Building and Engine Co., Chester, Pa., for the Old Dominion Line.

The length will be about 320 feet; beam, 42 feet; and depth of hold, 28 feet. The vessel is to be delivered by August 1, 1897, with a bonus of $100 for every day before, and an equal penalty for every day after that date.

The single engine will be of the triple expansion type with cylinders 30, 48 and 75 inches in diameter, the stroke being 56 inches. There will be four boilers.

The cost of the vessel is estimated at about $350,000.

Bremen and **Friedrich der Grosse.**—These large freight steamers for the North German Lloyd Steamship Company were described on page 861 of Volume VIII. The displacement of the *Bremen* is given as 18,000 tons on a draught of 28 feet.

The *Friedrich der Grosse* made her trial during the last week in November, 1896, the maximum I.H.P. being about 7,100, and the mean speed, 16 knots.

Kaiser Wilhelm der Grosse.—The JOURNAL has been informed that the statement made on page 861 of Volume VIII, that the feed, fire, bilge and water service pumps of this steamer are of the Blake vertical type, should have been "auxiliary feed pumps, &c." The main feed pumps are of the Weir type, supplied by Messrs. G. & J. Weir, Glasgow. The Blake Company furnished nine pumps besides the two main air pumps, and including the pumps for the ash ejectors.

Leinster and **Munster.**—These twin screw, steel steamers, built by Messrs. Laird Bros., Birkenhead, for the City of Dublin Steam Packet Co., were successfully put into the water on September 12 and October 21, 1896, respectively. The *Munster* was floated out of dock. These vessels are sisters of the *Ulster*, fully described on page 657 of Volume VIII. Their designed speed is 23 knots.

India.—The principal data of this single screw passenger steamer of the Peninsular and Oriental Co. were given on page 421 of Volume VIII. The following additional particulars are from "The Engineer," London.

The vessel is built of steel and classed at Lloyd's as 100 A 1. There are five principal decks, of which the upper, main and lower decks extend from stem to stern. The upper deck beams are Butterly bulbs, 10 inches deep, while those of the lower and orlop decks, of the same kind, are 12 inches deep. The double bottom extends the whole length of the vessel, and is 50 inches deep along the keel line. There are eleven athwartship bulkheads, of which nine are solid to the upper deck, the other two having water tight doors. All bulkheads have been built to meet the Admiralty requirements for auxiliary cruisers.

There are accommodations for 320 first and 160 second class

passengers, besides the ship's complement, which consists of from 450 to 500 officers and men.

Except in the engine room, where three upcast and one downcast ventilators are fitted with Schiele fans, no mechanical means have been adopted for ventilation. But every possible method of securing ventilation by natural means has been adopted,—large air ports, deck skylights and ventilators, and wide, unbroken, fore-and-aft passages.

The triple expansion, four crank engine, designed to develop nearly 10,000 I.H.P., has cylinders 42.5, 68 and (2) 74.5 inches in diameter, the stroke being 72 inches. The crank shaft is made in interchangeable parts, and is, like the rest of the shafting, of Vickers steel. The thrust blocks are of the solid ring pattern. The propeller is four-bladed, and of manganese bronze.

There are three double and three single-ended steel boilers, all 14 feet 10 inches in diameter. The working pressure is 170 pounds per square inch. The double-ended boilers have six, and the single-ended, three 42-inch furnaces each. There are two smoke pipes. Howden's forced draft system is used, three 8-foot Chandler fans, each worked by a compound, single crank, single-acting engine, supplying the air. The fans can deliver about 50,000 cubic feet of air each, with about 250 revolutions per minute.

There are two athwartship bunkers, and side bunkers along the length of the machinery compartments above the lower deck. In addition, there is a large reserve hold forward of the forward bunker, which can be used for coal or cargo.

A Weir's direct contact feed water heater is fitted. There are two Weir feed pumps, each with a cylinder 12.25 inches in diameter and a stroke of 26 inches; each of these pumps is capable of feeding the boilers at full power. A Weir's evaporator, using steam from the I.P. receiver, and with a capacity of 30 tons of fresh water per day, is installed. For potable water, one of Caird & Raynor's distillers is fitted.

The electric lighting is on the single wire system, the return circuit being through the hull. There are about 780 16-candle power lamps and a search light. There are three sets of Tangye,

vertical, 40-H.P. engines, with cylinders 8 and 16 inches in diameter, and a stroke of 10 inches. Each engine is direct-coupled to a Siemens H.B. $\frac{16}{20}$ dynamo, capable of supplying 220 amperes at 105 volts, with 190 revolutions per minute of the engine.

The anchor hoist is worked by steam. The eight hoisting cranes, with lifting powers ranging from 20 to 30 hundredweight, the winches, and two warping capstans, are worked by direct-acting hydraulic engines. A direct-acting, vertical, compound steam pump in the engine room supplies the pressure. The accumulator consists of a series of steel pipes, about 13 inches in diameter and 13 feet long, containing compressed air. The air compressor is small, and the engine has its own surface condenser.

The refrigerating machinery, in the after end of the engine room, consists of a compound duplex machine, made by the Haslam Foundry and Engineering Co., Derby, capable of discharging 80,000 cubic feet of air at a temperature ranging from 50 to 80 degrees Fahrenheit below zero. Although a duplex machine, either side can be worked separately. There are two steam cylinders, two air compressors, two expansion cylinders and two water pumps. The steam cylinders are compound, the working pressure being 160 pounds per square inch. A surface condenser is contained in the bed plate of the machine, and air, circulating and feed pumps are secured to the bed plate. Besides these, there is a water tower, for washing the air with a spray of water, coupled to a set of patent copper drying pipes.

Satisfactory trials of the *India* were made in the Firth of Clyde. On her first voyage, a speed of 18.33 knots was maintained for a considerable time.

Milwaukee.—This large, single screw, cargo steamer, built by Messrs. C. S. Swan & Hunter, Wallsend-on-Tyne, for Messrs. Elder, Dempster & Co., for the cargo and cattle trade from America to European ports, was successfully launched on November 7, 1896.

The length over all is 484 feet, between perpendiculars, 470 feet; beam, 56 feet; depth, moulded, to the awning deck, 42.25 feet. The dead weight cargo will be 11,500 tons, and the measurement cargo, over 18,000 tons, in addition to about 700 tons

of permanent bunker capacity. She is built to the highest class at Lloyd's on their three-deck rules, with scantlings in many instances considerably above the rule requirements, so as to stand heavy Atlantic weather. The upper deck is completely flush with the exception of steel houses midships, in which the officers will be accommodated, and a steel steering house aft. The crew will be berthed forward in the topgallant forecastle. As cattle will be carried, there is fitted between decks a complete outfit of cattle stalls, water service, etc. For rapidly handling the enormous cargoes, a large number of hatches, with 12 powerful steam winches and side derricks, with ample and powerful gear, are supplied.

The triple expansion engine, built by the North Eastern Marine Engineering Company, Wallsend, has cylinders 28, 46 and 75 inches in diameter by 54 inches stroke. Three large boilers supply steam at 180 pounds pressure, and are fitted with Howden's system of forced draft. The propeller blades are of manganese bronze. Davis & Co.'s steam steering gear and See's ash ejectors are fitted.

On January 16, the trial trip was made, the vessel being light and having on board about 1,800 tons of coal and 2,500 tons of water ballast. The I.H.P. was about 3,500 and the speed, 12 knots.

Avondale Castle and **Tintagel Castle.**—These steel steamers, built by the Fairfield Shipbuilding and Engineering Co., Govan, for the Castle Mail Packet Company, are practically sister ships. The *Tintagel Castle* was launched September 12, 1896, and had her trials on November 17. The other vessel was launched on November 5. The following description of one of these vessels is from "Engineering," London.

The length is 440 feet; beam, 50 feet; depth, 33 feet; gross tonnage, about 5,600 tons. The vessel is divided into numerous water tight compartments, and there is a double bottom running the whole length, which again is subdivided into various compartments for carrying water ballast. The cargo holds are five in number, and have capacity for about 6,000 tons of cargo, and each hold has a separate hatch and double steam winches for quick despatch in loading and discharging. The main hatch is specially constructed to take in boilers and mining machinery of

the largest dimensions, for lifting which a steel spar is provided, with adequate gins and steel wire gear. Four steel masts support the cargo gear, and provide for an ample spread of canvas. Steel yards and square sails are carried on the foremast.

The passenger accommodation, including about 170 first and second class berths, is placed amidships, with dining saloons, drawing and smoking rooms, for each class. The engines are of the three cylinder triple expansion type, working at a high pressure, steam being supplied by four tubular steel boilers. The shafting is of forged ingot steel, and the propeller of bronze. The power developed will insure a regular speed to South Africa of about 13 knots.

On the trials, the vessels being in a light condition, a speed of 14.75 knots was easily maintained by the *Tintagel Castle* and about 15 knots by the *Avondale Castle*.

Dunolly Castle.—This steel steamer, built by Messrs. Barclay, Curle & Co., Clydeholm, Whiteinch, for the Castle Mail Packet Co., was successfully launched on November 4, 1896.

The length is 382 feet; beam, 46.25 feet; depth, 30 feet; gross tonnage, about 4,150 tons. Saloon and state room accommodation is provided under the bridge deck for thirty first class passengers, and many emigrants may be carried, while at the same time extensive room is left for cargo.

The machinery was built at the Finniston shops of the above firm. The triple expansion engines have cylinders 26.5, 44 and 70 inches in diameter, the stroke being 48 inches. There are three single-ended boilers, the working pressure being 180 pounds per square inch.

At the luncheon which followed the launch, Sir John Muir stated that if the order for this vessel had been placed now instead of a year ago, the price would have been about $250,000 greater.

On January 19, a speed of 13.5 knots was attained on the official trial.

Kanagawa Maru, Kamakura Maru and **Hakata Maru.**—These steel, twin screw steamers, the first of ten building in Great Britain for the Nippon Yusen Kaisha, were successfully launched on October 22, December 9 and 21, 1896, respectively. The hulls are practically the same, and the machinery, identical.

The *Kanagawa* and *Hakata* were built by Messrs. D. and W. Henderson & Co., Glasgow, and the *Kamakura*, by Messrs. Workman, Clark & Co., Belfast. The following description of the *Kanagawa* is from the "Engineers' Gazette," London.

The length over all is 463 feet; beam moulded, 49 feet 2 inches; depth moulded, 33 feet 6 inches; and gross tonnage, 5,824. Being intended for passengers as well as cargo, the vessel has fairly fine ends, which will ensure her being a comfortable sea boat, and at the same time carry a large deadweight, and attain an average sea speed of fully 12 knots. There are nine watertight bulkheads, and a complete double bottom fore and aft, also well subdivided; in addition, there is a deep hold tank abaft the engine space extending to the height of the orlop deck, which can be utilized for cargo or water ballast. The main and upper decks are of steel, the upper deck being sheathed with teak. The steam steering gear is of Caldwell and Co.'s make. The machinery will be supplied by the builders' firm, and consists of two sets of triple expansion engines, each having cylinders 20, 33½ and 56 inches in diameter by 48 inches stroke. Steam is supplied by two large double-ended and two single-ended boilers, constructed for a working pressure of 200 pounds per square inch. The propellers are of manganese bronze. Among the auxiliary gear there are Weir's feed heater pumps and evaporator and Murdoch's governor.

On December 10, 11 and 12, a series of successful trial trips were made; a mean speed of over 14 knots was easily attained.

Princess Clementine.—This steel, paddle wheel, channel steamer, built by the Cockerill Co., Hoboken, near Antwerp, for the Belgian Government for the Dover-Ostend route, was successfully launched in November, 1896. This vessel completes the fleet of nine modern fast steamers for this service.

The length over all is 351 feet, on water line, 339.5 feet; beam, inside of paddle boxes, 37 feet 10 inches, over paddle boxes, 77 feet 7 inches; depth of hold, 15 feet; depth, from keel to upper deck, 23 feet 2 inches. There are three decks, and the internal fittings are very fine. The compound engines are designed to develop 8,500 I.H.P. and give the vessel a speed of 22.5 knots. There will be eight cylindrical boilers, fitted with Serve tubes.

YACHTS.

Marietta.—This steel steam yacht is building by the John N. Robins Co. at the Erie Basin Dry Docks, Brooklyn, N. Y., for Mr. Harrison B. Moore of the same city. The vessel is to be finished by June 1. The following information has been kindly supplied by the builders of the hull and by The Roberts Safety Water Tube Boiler Co., the builders of the boilers.

The vessel was designed by Mr. Henry J. Gielow, and will be built of open hearth steel, the latter to have a tensile strength of from 55,000 to 60,000 pounds per square inch, and an elongation of not less than 25 per cent. in eight inches. The hull will be divided by two main bulkheads, besides the collision bulkhead forward, and will have very sharp lines. The length over all will be 172.5 feet, and on the water line, 140 feet; beam, moulded, 18 feet; depth, amidships, 10 feet; displacement, 125 tons at a draught of 5 feet 7.5 inches. There will be one short deck house forward, two hollow masts for steadying sails and one smoke pipe.

The engine is of the four cylinder, triple expansion type and is building by Mr. J. W. Sullivan, New York. The cylinders will be 14, 21.5 and (2) 25 inches in diameter and the stroke, 18 inches. The designed I.H.P. is 1,000 at 300 revolutions per minute, and a working pressure in the boilers of 250 pounds per square inch. The engine will be forward of the boilers, the shaft running between the boilers.

There will be two Roberts water tube boilers, placed side by side. Each boiler will be approximately 6 feet 3 inches wide, 9 feet 9 inches long and 6 feet 8 inches high, exclusive of the depth of the ash pan. The latter will probably be built to fit the frames of the vessel. The total grate surface will be 80, and the heating surface, about 2,700 square feet. The estimated weight of the two boilers will be about 26,000 pounds. Forced draft on the closed fire room system will be used.

Hiawatha.—This steel yacht is building by the consolidated firms of the Gas Engine and Power Co. and Messrs. C. L. Seabury & Co., at Morris Heights, New York City, for Mr. Chas. Fleischmann, who found that the old *Hiawatha* built last year, and described on page 214 of Volume VIII, was too small.

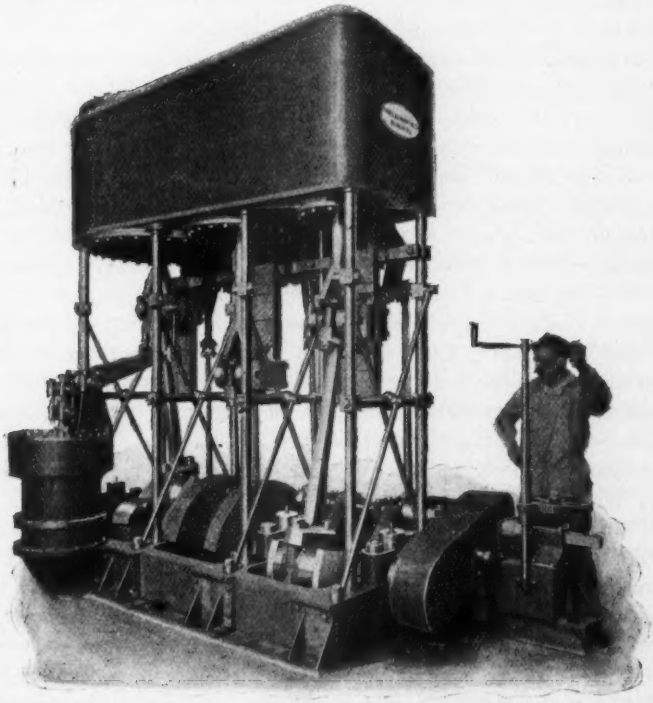
The following particulars have been kindly supplied by the builders, to whom the JOURNAL is also indebted for the cut of the engine.

The length of the new *Hiawatha* will be 170 feet over all and 136 feet on water line; beam, 21 feet; and draught, 8 feet. The hull will be of steel, the decks of white pine, and the bulwarks capped with polished teak. The yacht will be schooner rigged, and have one smoke pipe.

There will be two deck houses, a large one forward, and a smaller one, aft. The forward deck house will be finished in mahogany inside and outside, and fitted with a buffet and large dining table, butler's pantry, and dumb waiter leading to the galley below. There will be one state room in the after part of this house. The after deck house will also be finished in mahogany, and fitted as a lounging room. A companionway on the starboard leads to the saloon and rooms below. On the lower deck forward will be the chain locker, windlass, and the officer's and crew's quarters, the latter to be finished in butternut. There will be berths for twelve men, and state rooms for the captain, engineer, mate and steward. Opposite the latter will be the officers' mess room. Abaft of this will be the galley, fitted with a large range and all modern conveniences. Still further aft, and immediately forward of the machinery space, will be two large state rooms for guests, reached by a companionway from the main deck. These will be finished in white and gold, like the other state rooms, and be fitted with brass bedsteads, dressing cases and toilet conveniences.

Abaft of these rooms will be a large bunker, the athwartship fire room, two boilers, and then the engine room. Two large state rooms for the owner's use come next, fitted with beds, clothes' lockers, wardrobe, bath, etc. Abaft of these will be two more state rooms, separated by a passage which leads to the main saloon aft. This room will be finished in polished

mahogany, and have a large circular center table, writing desk and piano. The companionway to the after deck house will lead out of this on the starboard side aft, and a passageway on the port side, to one single and one double state room. On the starboard side, next to the companionway, will be a linen closet and toilet.



The engine will be of the triple expansion type, to be built by Messrs. C. L. Seabury & Co., and is expected to develop about 1,000 I.H.P., giving the yacht a guaranteed speed of 18 miles per hour, or about 15.5 knots. The cylinders will be 13.5, 21 and 34 inches in diameter, the stroke being 21 inches.

There will be two Seabury water tube boilers of the latest design, each to have 50 square feet of grate surface, and 1,874 square feet of heating surface. The working pressure will be 260 pounds per square inch.

The yacht will be heated by steam and lighted by electricity. She will be fitted with a steam capstan, and carry one 25-foot naphtha launch, one 20-foot gig, a 16-foot cutter, all mahogany-finished, and a 12-foot dinghy, oak-finished.

Varuna.—This steel, twin screw yacht was built by Messrs. A. & J. Inglis, Pointhouse, Scotland, for Mr. Eugene Higgins of New York, from designs by Mr. G. L. Watson. The gross tonnage is about 1,500 tons, and the designed speed, 16.5 knots. The launch took place on July 14, and the trials, in November, 1896. The length over all is 310 feet, and on the water line, 260 feet; beam, molded, 35 feet 2 inches; depth, 18 feet 4 inches.

The engines are of the four cylinder, triple expansion type, with cylinders 22.5, 38 and (2) 40 inches in diameter, the stroke being 27 inches. The number of revolutions with the boiler working pressure is 165 per minute. There are two single-ended boilers, each 17.5 feet in diameter, and about 12 feet long, with four very large corrugated furnaces in each. The working pressure is 160 pounds per square inch. The shell plates are 1⅛ inches thick.

Progressive trials were satisfactory, the speed, with natural draft, being 14.5 knots. On November 12, the full power trial was made in the Firth of Clyde. Two runs each way were made between the Cloch and the Cumbrae, the mean speed being between 16.75 and 17 knots. A 24-hours' coal consumption trial was made on the next day, but no results have been published. A few days later the yacht left for New York where she arrived safely.

Mayflower.—This steel, twin screw yacht was built by Messrs. J. & G. Thomson, Clydebank, for Mr. Ogden Goelet, from designs by Mr. G. L. Watson. She is larger and has more boiler power than the *Varuna*, described above, but the engines are of the same size. She was launched on November 7, 1896.

The length over all is 320 feet, and on the water line, 275 feet; beam, moulded, 36.5 feet; depth, 21 feet; mean draught, about 16.5 feet; and gross tonnage, 1,800. The rig is that of a two-masted schooner. The engines are of the four cylinder, triple expansion type with cylinders 22.5, 38 and (2) 40 inches in diameter, the stroke being 27 inches. There are two double-ended, cylindrical boilers which may be worked under forced draft.

The electric light plant consists of duplicate sets of dynamos and a battery of accumulators. Two search lights and refrigerating machinery are provided. The complement consists of about 80 men.

———.—This steel yacht is building by the Harlan & Hollingsworth Co., Wilmington, Del., for Mr. G. W. Childs Drexel of Philadelphia. The designs are by Mr. A. S. Chesebrough, Boston.

The length on the water line will be 150 feet 8 inches, and on deck, 170 feet 8 inches; beam, 24 feet; and draught, 14.75 feet.

The triple expansion engine will have cylinders 14, 22 and 36 inches in diameter, the stroke being 22 inches. The propeller will be of manganese bronze. A search light will be provided.

Marietta II.—The following additional particulars of the machinery of this British-built yacht, described on pages 660 and 873 of Vol. VIII, are from the "Marine Review," Cleveland.

The twin screw, quadruple expansion engines have cylinders 15.5, 22, 31 and 44 inches in diameter, the stroke being 27 inches. The bunker capacity is 380 tons, sufficient for 15 days at a speed of 12 knots. All the auxiliary machinery is run by steam.

Christine.—This yacht, built by Messrs. D. J. Dunlop & Co., Port Glasgow, for Sir Wm. Scott, was launched on December 30, 1896, and made her official trial on January 13. The following particulars are from "Engineering," London.

Length over all, 150.5 feet, and between perpendiculars, 130 feet; beam, 20 feet; depth moulded, 12.5 feet; net tonnage, 107.44, and gross tonnage, 195.59. Tonnage by Thames yacht measurement, 245 tons.

The triple expansion engine has cylinders 11, 18 and 28 inches

in diameter, the stroke being 24 inches. There is one single-ended boiler, the working pressure being 160 pounds per square inch.

The trial was satisfactorily made on January 13, the means of two runs over the Skelmorlie mile giving a speed of 11.4 knots.

Eugenie.—This single screw yacht, built by Messrs. Hawthorne & Co., Leith, Scotland, for Mr. J. G. Cassatt of Philadelphia, was launched on November 5, 1896. The gross tonnage is 157 tons. The triple expansion has cylinders 11.25, 18 and 30 inches in diameter. The working pressure will be 160 pounds per square inch.

Ivy.—The part to be played in the Benin Punitive Expedition by this steam yacht suggests a brief description, taken from "Engineering," London. She was built by the Earle Company at Hull, and handed over to the Protectorate in less than a year from the time of commencement. Her ordinary speed of 10 knots can be increased to 13 knots; she is very roomy, an excellent sea boat, with plenty of accommodation for her officers and crew, besides 10 spare cabins and a large troop deck for the transport of military; she can carry about 500 native soldiers. Her dimensions are: Length over all, 237 feet, and 204 feet between perpendiculars; breadth, 34 feet; and depth, 16.5 feet. She is constructed on the composite principle. The deck house is 130 feet in length and 20 feet in breadth, built of teak, as are also all the cabin fittings and furniture. The yacht is specially ventilated. She carries a 7-pounder forward and two machine guns aft. The machinery consists of two sets of triple expansion engines of 1,150 I.H.P., which drive a pair of bronze propellers, and steam is supplied from two large steel multitubular boilers working at 150 pounds pressure. Her draught of 10 feet 3 inches enables her to get further up the Niger than any of H.M. ships on the station.

BOOK REVIEW.

THE STEAM NAVY OF THE UNITED STATES.—It has been reserved for an officer of the Naval Engineer Corps, Passed Assistant Engineer Frank M. Bennett, U. S. N., to produce a new form of naval history, differing from all its predecessors in that it acknowledges the presence of steam machinery on board ship and deals with the tremendous influence steam has exerted upon naval events during the last half century. His book, published by Warren & Co., Pittsburg, Pa., is an octavo of over 900 pages, with many fine illustrations, and is what it purports to be, "A history of the growth of the steam vessel of war in the U. S. Navy, and of the Naval Engineer Corps."

It will surprise most well-informed people to learn from this volume that a steam war vessel was built in this country during the war of 1812, and that the vessel was a formidable one. This craft was designed and built under Government patronage by Robert Fulton, and was named by him *Demologos*, but was entered on the navy list as *Fulton*. Her length was 156 feet; breadth, 56 feet, and displacement about 2,500 tons. The motive power was a single large engine and boiler driving a paddle wheel located about the middle of the vessel, which had practically two hulls to accommodate this arrangement. Twenty heavy guns were mounted on the upper deck. The war with Great Britain having ended before this pioneer war steamer was completed, the opportunity of testing her in battle failed. During the summer of 1815 she had some entirely successful steam trials, and was then laid up at the Brooklyn Navy Yard as a receiving ship. In 1829 her magazine blew up, killing and wounding a great number of people, and completely destroying the vessel. This *Fulton* is famous as the first war steamer ever built, her construction antedating by many years the application of steam to naval purposes in any other country.

In 1835, the Secretary of the Navy discovered that Congress in 1816 had authorized the building of a war steamer, and he ordered the board of navy commissioners forthwith to prepare plans and proceed with its construction. This ship was named *Fulton*, in honor of the great engineer, and is known in naval records as "*Fulton the Second*." She was built at the New York Navy Yard and completed in 1837; was 180 feet long, 35 feet beam, of about 1,200 tons displacement, and steamed at the rate of 12 knots. The side wheels were worked by two horizontal condensing engines of 625 horse power. The cost was about $300,000, or almost exactly the original cost of the frigate *Constitution*. Her captain, Mathew C. Perry, reported that she was "more useful than any number of armed ships not propelled by steam." Our author praises him highly as the father of the steam navy, and the title seems deserved from the account of his great interest in marine engines and engineers.

The development of steam for commercial purposes at sea was going forward rapidly and foreign nations were applying it to their war vessels; so, when the "*Fulton the Second*" proved so successful, there was a public demand for more war steamers. This was not without opposition; although the military superiority of a self-moved battery over one dependent upon the whims of the wind is obvious.

In 1839, work was begun on the side wheel steamers, the *Mississippi* and *Missouri*, and completed in 1842. They were large for the time, 229 feet long, 40 feet beam, and of 3,220 tons displacement. The former had side lever, and the latter, direct-acting engines, and both had copper boilers of the return ascending flue variety. In 1843 the *Missouri* was burned to the water's edge in the harbor of Gibraltar. The *Mississippi* had a long and famous career and participated in many historical events.

Contemporary with these two ships was the side wheel gunboat *Michigan*, still in service. Built of iron at Pittsburg in 1842, the hull plates, machinery, etc., were carried overland and assembled at Erie, where the completed ship was launched in 1843. She was the first iron vessel in the Navy, and was the first iron

steamer afloat on the great Lakes. Her original engines are still in use, though the boilers have been renewed.

At this same period, we had some novel experiments with submerged propulsion in the Navy. A naval lieutenant named Hunter conceived the idea of turning side wheels half way over so they would be horizontal and below the surface of the water. The object to be gained was desirable, for the usual form of side wheel was exposed to shot and filled in considerable space that might be used for broadside guns. Three vessels were built on his plans, but all were failures.

At the same time John Ericsson was engaged on the *Princeton*, the first screw-propelled war vessel ever built. So important is this event, that Mr. Bennett gives a whole chapter to this vessel. It was upon the *Princeton* that the explosion of the great gun occurred, by which the Secretary of the Navy, Secretary of State and several other distinguished guests were killed.

The superiority of steam over sails was so well proved during the War with Mexico that serious efforts were made to build a fleet of war steamers. The first fruits were the side wheel steamers *Powhatan* and *Susquehanna*—larger and improved models of the *Mississippi*—and the ill-fated *San Jacinto*, a screw propeller. Then came (in 1855) those grand steam frigates like the *Merrimac* and *Wabash* and the big steam sloop of war *Niagara*, which laid the first Atlantic cable. Two years later the *Hartford* class of steam sloops was added, and closely following came the four of the *Dacotah* class and the three of the *Narragansett* class. All this building progressed during the fifties, and by the time the great civil war burst upon us, the American navy had a respectable fleet of war steamers. They are described in detail with many illustrations and the text is fascinating.

The engineer to the board of navy commissioners while the second *Fulton* was building was Mr. Chas. H. Haswell of New York. He soon after received the appointment of chief engineer of the *Fulton*, and in 1844 became engineer-in-chief of the Navy. When appointed to the *Fulton*, he was the first, and for a time, the only engineer in the Navy. As Mr. Haswell is yet living, it

has been his rare privilege to see and take part in the development of steam navigation from its early rude examples to the present perfected form.

An interesting chapter is devoted to a historical review of the changes in the appointment and training of the officers of the Engineer Corps.

Sixteen chapters are devoted to naval operations and to the remarkable development of war ship construction during the civil war. Particular attention is given to the monitors and other armored vessels and to the swift cruisers brought into existence by war requirements. The responsibility of engineer officers in that war is fully shown, and their share of duty is attested by the long list of killed and wounded.

Perhaps the most absorbing chapter is one devoted to arctic exploration, in which the voyages of the *Polaris*, *Jeannette* and *Rodgers* are told again, and the wonderful story of the heroism and fortitude of Chief Engineer Geo. W. Melville appears in fresh form. An account is also given of the Greely Relief Expedition. One chapter of general interest treats of the impetus given to technical education by the detail of naval engineers as instructors in scientific colleges. The change in public opinion that led to the regeneration of our Navy and the constructive progress of the new fleet are exceedingly well and clearly described in the concluding chapters.

By collecting all this valuable historical matter and arranging it in orderly and interesting form, Mr. Bennett has reflected great credit upon himself as a student and has given a work of immense value to the Naval service, to historians, and to the general reading public. His work is of engrossing interest now, and in the future will be quoted as an authority upon what will be considered the most interesting and important transformation our Navy has experienced; for, as he truly says in his conclusion, "engineers, whether by that name or some other, but engineers nevertheless, will hold in their hands the fate of the coming years."

OBITUARY.

WILLIAM SCHARF SMITH.

Chief Engineer W. S. Smith was born in Baltimore, Md., April 23, 1836. He entered the Navy as a Third Assistant Engineer, February 21, 1861; was promoted to Second Assistant Engineer, January 16, 1863; to First Assistant Engineer, December 1, 1864; and was commissioned a Chief Engineer on October 20, 1875.

Of his whole naval service of about thirty-six years, seventeen years were sea service and thirteen years, shore duty.

His first sea duty was on the *Colorado* on the North Atlantic Station, and his last, on the *Lancaster*, June, 1894. Among the other vessels to which Mr. Smith was attached may be mentioned the *Nipsic* on the Darien expedition, 1869–1870.

He was on special duty in connection with the building of the *New Ironsides*, 1862–1863, and with the new crusiers building at the Union Iron Works.

He attained the relative rank of Commander on August 10, 1893, and was twice a member of the Board of Examining Engineers at Philadelphia, being on the latter duty at the time of his death.

Mr. Smith died in Philadelphia on February 7, 1897.

ANNUAL MEETING AND SPECIAL NOTICE.

The annual meeting of the Society was held on January 9, 1897, in the office of the Engineer-in-Chief of the Navy.

Chief Engineer A. B. Canaga, U. S. N., was elected chairman.

There being no papers for discussion before the Society, the report of the Council on the Prize Competition, and the report of the Secretary-Treasurer were read, the latter showing a solid financial condition. A statement was also made showing a gratifying increase in the membership of the Society. The reports are as follows:

REPORT OF THE COUNCIL.

WASHINGTON, *January 8, 1897.*

To the American Society of Naval Engineers:

Sirs: In accordance with the resolution adopted at the last annual meeting, January 11, 1896, the Council extended invitations to the Members and Associate Members of this Society to submit papers for competition.

Owing to the limited number of papers submitted, and to the fact that these were not considered up to the standard of the JOURNAL, the Council regrets that it could not award any prizes.

The Council respectfully suggests that the invitation, changed as follows, be renewed for the coming year.

To the Members and Associate Members:

In accordance with the resolution adopted on January 11, 1896, at the annual meeting of the Society of American Naval Engineers, the Council takes pleasure in offering for competition, according to the rules of the resolution (printed on page 221 of Volume VIII), the sums of $75 and $25 respectively, and suitable medals which will be prepared, for the two best, original papers. The Council will award "Honorable Mention" to such other papers as may deserve this distinction. The Council reserves the right to reject any or all papers.

By the broad terms of the resolution, the subject of a paper may be in the form of an original essay on any matter suitable for this Society, or it may be on the results of original experiment or research. There is no limit to the length of the papers, nor in the number of illustrations which may be thought necessary.

The Council expresses the hope that members and associate members will strive for the honors thus bestowed by the Society, and by their original productions enhance the growing value of the JOURNAL.

Respectfully submitted, by direction of the Council,

EMIL THEISS,
P. A. Engineer, U. S. N.

REPORT OF THE SECRETARY-TREASURER.

WASHINGTON, *January 9, 1897.*

To the American Society of Naval Engineers:

Sirs: In accordance with the requirements of Section 14 of the By-Laws of the Society, I have the honor to submit the following report on the financial condition of the Society from January 11, 1896, to date.

RECEIPTS.

Balance on hand, January 11, 1896	$2,815.85
Annual dues	2,631.90
Subscriptions, sales and reprints	912.51
Advertisements	1,060.12
Interest on deposit	66.17
	$7,486.55

EXPENDITURES.

Printing of JOURNAL	$2,121.49
Illustrations	347.14
Salary of Secretary-Treasurer	600.00
Stationery, postage and incidentals	115.71
Purchase of JOURNAL	20.00
	$3,204.34
Balance on hand, January 9, 1897	$4,282.21

This balance is on deposit with the American Security and Trust Company.

Very respectfully,

F. C. BIEG, *Secretary-Treasurer.*

After the reading of the reports, which were unanimously approved, the election of officers was held, with the following results:

President: Chief Engineer David Smith, U. S. N., (retired.)

Secretary-Treasurer: Passed Assistant Engineer F. C. Bieg, U. S. N.

Members of Council, in addition to the President and Secretary-Treasurer:

Chief Engineer A. B. Canaga, U. S. N.; Passed Assistant Engineer H. P. Norton, U. S. N.; and Passed Assistant Engineer Emil Theiss, U. S. N.

A vote of thanks was then given to the Secretary-Treasurer for his efficient and successful management of the affairs of the Society.

It having been suggested that it would be desirable to add some social features, such as a dinner and reunion, to the annual meeting, P. A. Engineer F. H. Eldridge, U. S. N., moved that the Council be empowered to arrange for such an addition to the annual meeting, and to get the views of the members as to the time of holding the next annual meeting, and to invite suggestions from them on this subject.

This motion was unanimously adopted.

A motion made by P. A. Engineer R. S. Griffin, U. S. N., which was unanimously adopted, empowered the Council to get the views of the members on the adoption of a Society badge and button.

There being no further business, the meeting adjourned.

The report and accounts of the Secretary-Treasurer have since been audited and found to be correct.

SPECIAL NOTICE.

In accordance with the report adopted at the above annual meeting, the Council extends the invitation, printed above, to members and associate members to compete for the prizes offered. The resolution, referred to above, is here reprinted.

Resolved, That it is the sense of this meeting that the Council of the Society of American Naval Engineers should offer annually the sums of $75 and $25, and appropriate medals for the two best papers on any subjects suitable for the JOURNAL, and written by a member or associate member of this Society, subject to the following rules:

1. Papers must be original and submitted to the Secretary and Treasurer before the first of November of each year.

2. The Council of the Society shall judge the value of the papers submitted and award the prizes to the two best, and "honorable mention" to such other papers as deserve this.

3. Papers receiving the award of "honorable mention" shall be considered the property of the Society for publication in the JOURNAL.

4. All papers submitted shall bear a motto of the author, which motto shall be identified by a separate, sealed communication to the Secretary and Treasurer, giving the author's name and motto. These notes of identification shall be kept by the

Secretary and Treasurer, and opened only after the Council has given its judgment on the papers. Then the prizes and "honorable mention" shall be awarded by name and the papers published in the JOURNAL.

To carry out the resolutions regarding the addition of a dinner and reunion to the annual meeting, and the adoption of a badge and button, the Council cordially invites all members and associate members to send to the Secretary-Treasurer their views on these subjects. Designs for the badge or button are cordially invited.

F. C. BIEG,
Sec'y-Treasurer.

AMERICAN SOCIETY OF NAVAL ENGINEERS.

OFFICERS FOR 1897.

President:

Chief Engineer **David Smith**, U. S. Navy.

Secretary and Treasurer:

Passed Assistant Engineer **F. C. Bieg**, U. S. Navy.

Council:

Chief Engineer **David Smith**, U. S. Navy.
Chief Engineer **A. B. Canaga**, U. S. Navy.
Passed Assistant Engineer **F. C. Bieg**, U. S. Navy.
Passed Assistant Engineer **H. P. Norton**, U. S. Navy.
Passed Assistant Engineer **Emil Theiss**, U. S. Navy.

HONORARY MEMBERS.

(EX-OFFICIO.)

The Secretary of the Navy.

The Assistant Secretary of the Navy.

Chief of Bureau of Steam Engineering:

Geo. W. Melville, Engineer-in-Chief, U. S. N., Navy Department.

Ex-Chiefs of Bureau of Steam Engineering:

Chas. H. Haswell, Consulting and Superintending Engineer, 42 Broadway, New York.
B. F. Isherwood, Chief Engineer, U. S. N. (retired), 111 East 36th street, New York.
J. W. King, Chief Engineer, U. S. N. (retired), 3231 Powellton avenue, Philadelphia.
Chas. H. Loring, Chief Engineer, U. S. N. (retired), 239 Clermont avenue, Brooklyn.
Wm. H. Shock, Chief Engineer, U. S. N. (retired), 1404 15th street, Washington, D. C.

MEMBERS.

Able, A. H., Chief Engineer, U. S. N.....2034 Mt. Vernon street, Philadelphia, Pa.
Addicks, W. R., Chief Engineer Bay State, Boston and Brookline Gas Light Companies, 24 West street, Boston, Mass.
Aldrich, Wm. S., Professor Mechanical Engineering, and Director Department of Mechanic Arts, West Virginia University, Morgantown, W. Va.
Allderdice, W. H., Passed Assistant Engineer, U. S. N.....126 S. 19th street, Philadelphia, Pa.
Allen, D. Van H., Assistant Engineer U. S. N.....U. S. S. *Yantic*.
Allen, F. B., Vice-Pres. Hartford Steam Boiler Insp. and Ins. Co., res., 61 Willard street, Hartford, Conn.
Anderson, M. A., Passed Assistant Engineer, U. S. N., Inspector, Herreshoff Mfg. Co., Bristol, R. I.
Andrade, Cipriano, Chief Engineer, U. S. N., 273 Lafayette st., W. Germantown, Philadelphia.
Aston, Ralph, Chief Engineer, U. S. N.....76 Hanson Place, Brooklyn, N. Y.
Ayres, S. L. P., Chief Engineer, U. S. N., Examining Board, P. O. Building, Philadelphia, Pa.

Bailey, F. H., Chief Engineer, U. S. N.....U. S. S. *Newark*.
Baird, G. W., Chief Engineer, U. S. N., Superintendent State, War and Navy Dept. Bldg., Room 148, Navy Dept., Washington, D. C.

Ball, Walter, Assistant Engineer, U. S. N.....U. S. S. *New York.*
Barnard, G. A., Mechanical Engineer.....39-41 Cortlandt street, New York.
Barrett, Thos. H., late Assistant Engineer, U. S. N...Room 156, Post Office Building, New York.
Barry, J J., Passed Assistant Engineer, U. S. N. (retired),
106 McDonough street, Brooklyn, N. Y.
Bartlett, F. W., Passed Assistant Engineer, U. S. N.....U. S. S. *Vesuvius.*
Barton, J. K., Chief Engineer, U. S. N .....U. S. S. *Bennington.*
Bates, A. B., Chief Engineer, U. S. N.....U. S. S. *Franklin.*
Baughman, H C., Assistant Engineer, U. S. N. (retired), 2000 N. 12th street, Philadelphia, Pa.
Baxter, W. J., Naval Constructor, U. S. N.....Navy Yard, Mare Island, Cal.
Bayley, W. B., Chief Engineer, U. S. N.,
Bureau Steam Engineering, Navy Department, Washington, D C.
Beach, E. L., Passed Assistant Engineer, U. S. N.....U. S. S. *Puritan.*
Bennett, F. M., Passed Assistant Engineer, U. S. N.....U. S. S. *Amphitrite.*
Bevington, Martin, Passed Assistant Engineer, U. S. N.....U. S. S. *Columbia.*
Bieg, F. C., Passed Assistant Engineer U. S. N.,
Bureau Steam Engineering, Navy Department, Washington, D. C.
Borthwick, J. L. D., Chief Engineer, U. S. N. (retired).....317 West 6th street, Erie, Pa.
Bowers, F. C., Passed Assistant Engineer, U. S. N .....U. S. S. *Maine.*
Boyd, Jas. T., Consulting Engineer.....28 State street, Boston, Mass.
Brady, John R., Assistant Engineer, U. S. N..... U. S. S. *Indiana.*
Bray, Chas. D., Professor Civil and Mechanical Engineering.....Tufts College, Mass.
Brecht, T. C......Bureau Steam Engineering, Navy Department, Washington, D. C.
Brooks, W. B., Chief Engineer, U. S. N. (retired).....437 West 6th street, Erie, Pa.
Bryan, B. C., Passed Assistant Engineer, U. S. N.....U. S. S. *Dolphin.*
Buehler, W. G., Chief Engineer, U. S. N.....Navy Yard, Portsmouth, N. H.
Burd, G. W., Passed Assistant Engineer, U. S. N.....U. S. S. *Cincinnati.*
Burgdorff, T. F., Passed Assistant Engineer, U. S. N.....U. S. S. *Oregon.*
Burke, W. S., Passed Assistant Engineer, U. S. N.....27 Everett street, Cambridge, Mass.
Bush, W. W., Passed Assistant Engineer, U. S. N.....Moran Bros'. Co., Seattle, Wash.

Canaga, A. B., Chief Engineer, U. S. N.,
Bureau Steam Engineering, Navy Department, Washington, D. C.
Capps, W. L., Naval Constructor, U. S. N .....Union Iron Works, San Francisco, Cal.
Carney, R. E, Assistant Engineer, U. S. N. (retired).....P O. B. 236, Vallejo, Cal.
Carr, C. A., Passed Assistant Engineer, U. S. N.....U. S. S. *Marblehead.*
Carter, T. F., Passed Assistant Engineer, U. S. N.....U. S. S. *Brooklyn.*
Cathcart, W. L, Mechanical Engineer.....Gwynedd, Pa.
Chambers, W. H., Passed Assistant Engineer, U. S. N.....Navy Yard, New York.
Cleaver, H. T., Chief Engineer, U. S. N.....U. S. S. *Alert.*
Cline, H. H., Chief Engineer, U. S. N. (retired).....634 Greene ave., Brooklyn, N. Y.
Collins, Jno. W., Engineer-in-Chief Revenue Cutter Service,
Treasury Department, Washington, D. C.
Conant, F. H., Passed Assistant Engineer, U. S. N.....Naval Academy, Annapolis, Md.
Cone, Hutch I., Assistant Engineer, U. S. N.....U. S. S *Philadelphia.*
Cook, Allen M., Assistant Engineer, U. S. N.....U. S. S. *New York.*
Cooley, Mortimer E., Professor Mechanical Engineering,
University of Michigan, Ann Arbor, Mich.
Cooper, I. T., Assistant Engineer, U. S. N.....U. S. S. *San Francisco.*
Cowie, George, Jr., Chief Engineer, U. S. N.....U. S. S. *Puritan.*
Cowles, W. Barnum, Constructing Engineer and Naval Architect,
223 Princeton street, Cleveland, Ohio.
Crawford, Robt., Passed Assistant Engineer, U. S. N. (retired),
Superintendent Williamson School, Williamson School Post Office, Delaware Co., Pa.
Creighton, W. H. P., Assistant Engineer, U. S. N. (retired), 1420 Toledano st., New Orleans, La.
Cunningham, Thomas Scott, late First Assistant Engineer, U. S. N..
B. 10, Produce Exchange, New York.

Danforth, Geo. W., Passed Assistant Engineer, U. S. N .....Navy Yard, New York.
Day. W. B, Passed Assistant Engineer, U. S. N.....Navy Yard, New York.
Denig, R. G., Chief Engineer, U. S. N.....U. S. Training Station, Newport, R. I.
Dismukes, D. E., Assistant Engineer, U. S. N.....Navy Yard, Norfolk, Va.
Dixon, A. F., Chief Engineer, U. S. N.....U. S. S. *Detroit.*
Doran, James S., Superintending Engineer International Navigation Co.,
307 Walnut street, Philadelphia, Pa.
Dowst, F. B, General Superintendent B. F. Sturtevant Co.....Jamaica Plain, Boston, Mass.
Dripps, W. A., Mechanical Engineer.....3324 Walnut street, Philadelphia, Pa.
Dungan, W. W., Chief Engineer, U. S. N.....Inspector, Cramp's Shipyard, Philadelphia, Pa.
Durand, W. F., Principal, School of Marine Construction.....Cornell University, Ithaca, N. Y.
Dunning, Wm B., Passed Assistant Engineer, U. S. N.....U. S. S. *Olympia.*
Dyson, Charles W, Passed Assistant Engineer, U. S N.,
U. S. Fish Commission, Washington, D. C.

Eaton, Wm. C., Chief Engineer, U. S. N.....Hamilton, Madison Co., N. Y.
Eckart, W. R., Mechanical Engineer.....3014 Clay street, San Francisco, Cal.
Edson, Jarvis B., Mechanical Engineer.....313 W. 74th street, New York.
Edwards, Jno. R., Chief Engineer, U. S. N.,
Bureau of Steam Engineering, Navy Department, Washington, D. C.
Eldridge, F. H, Passed Assistant Engineer, U. S. N.....Naval Academy, Annapolis, Md.
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Sibley Journal of Engineering.....Cornell University, Ithaca, N. Y.
Stevens Indicator.....Stevens Institute, Hoboken, N. J.

Technology Quarterly.....Massachusetts Institute of Technology, Boston, Mass.

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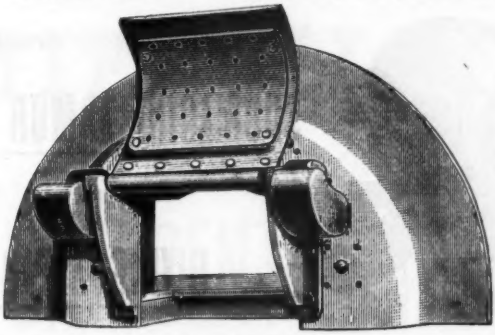

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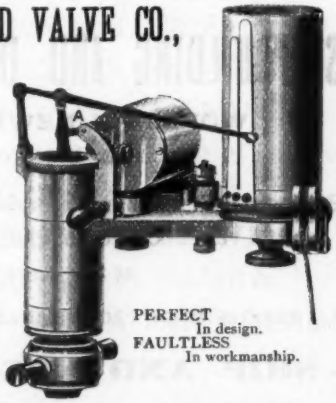

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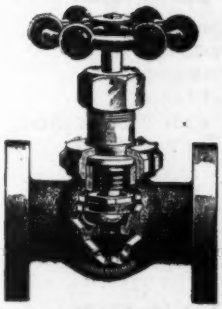

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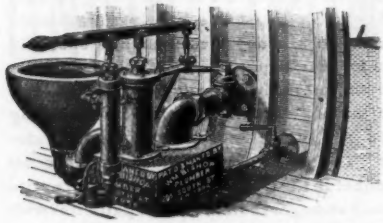

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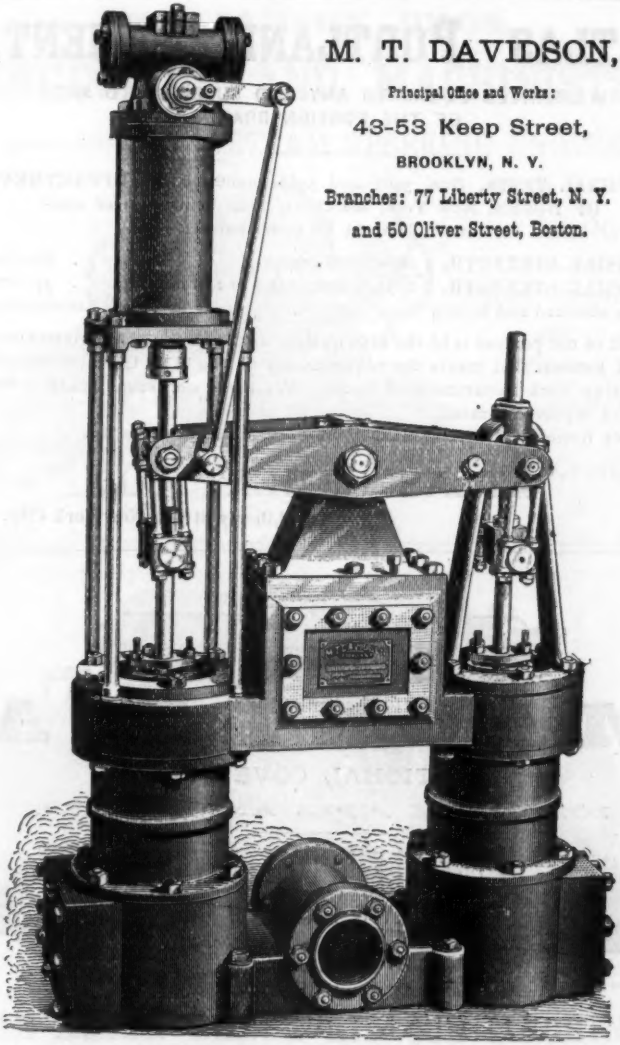

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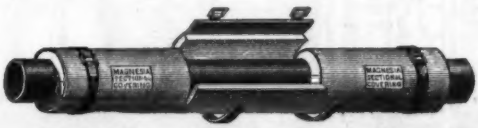

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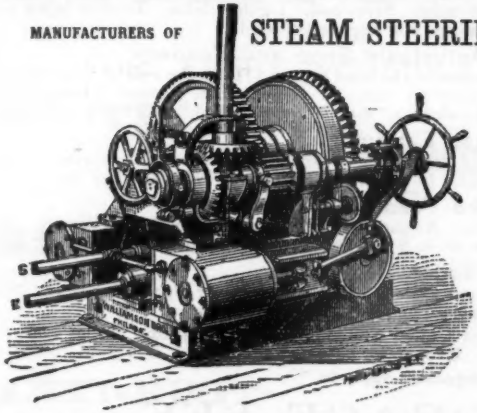

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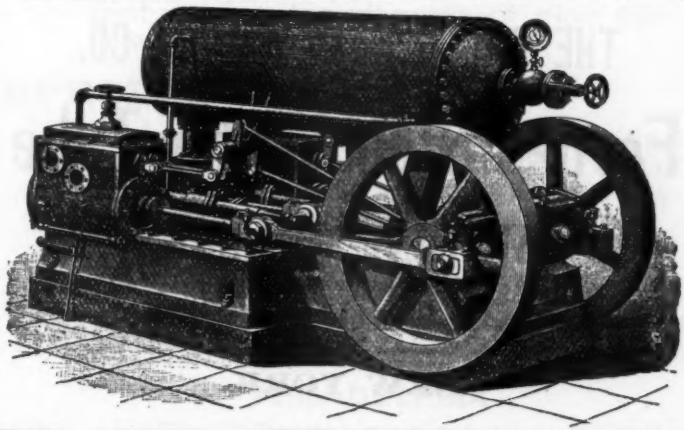

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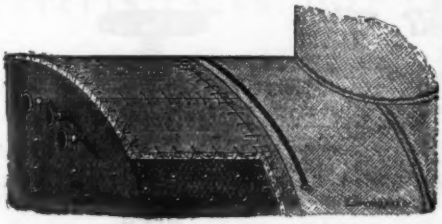

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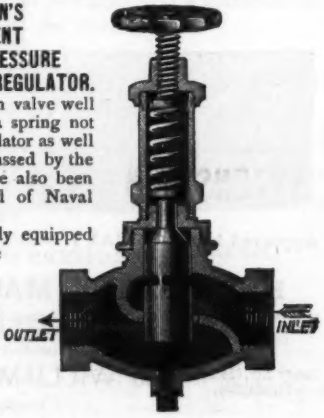

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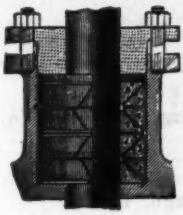

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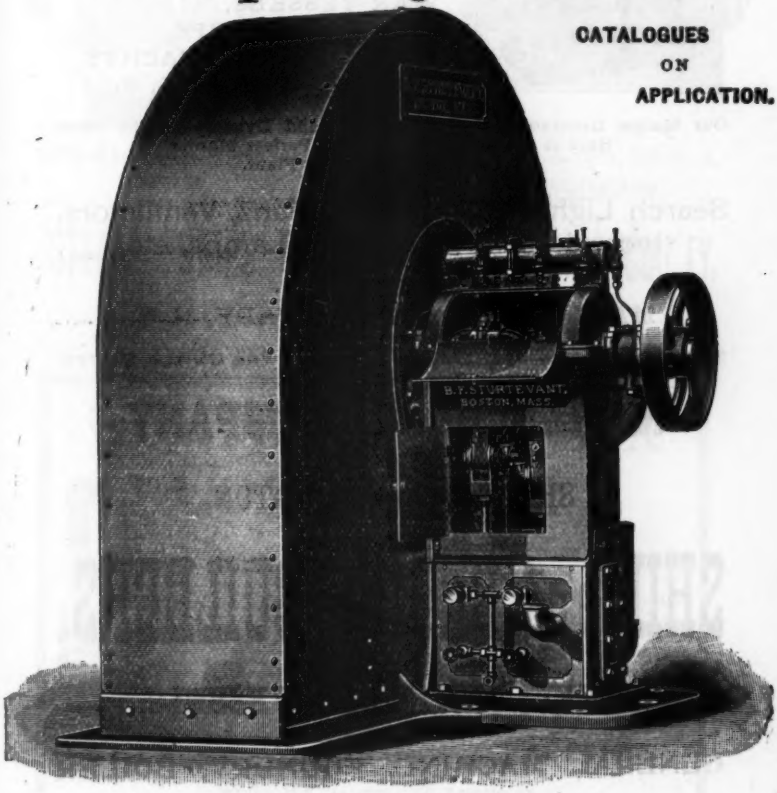

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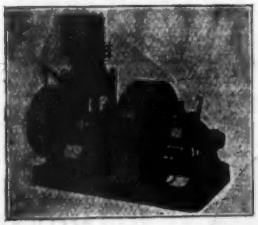

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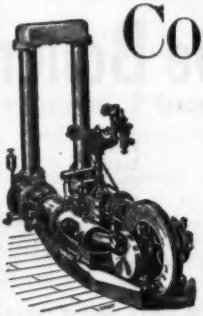

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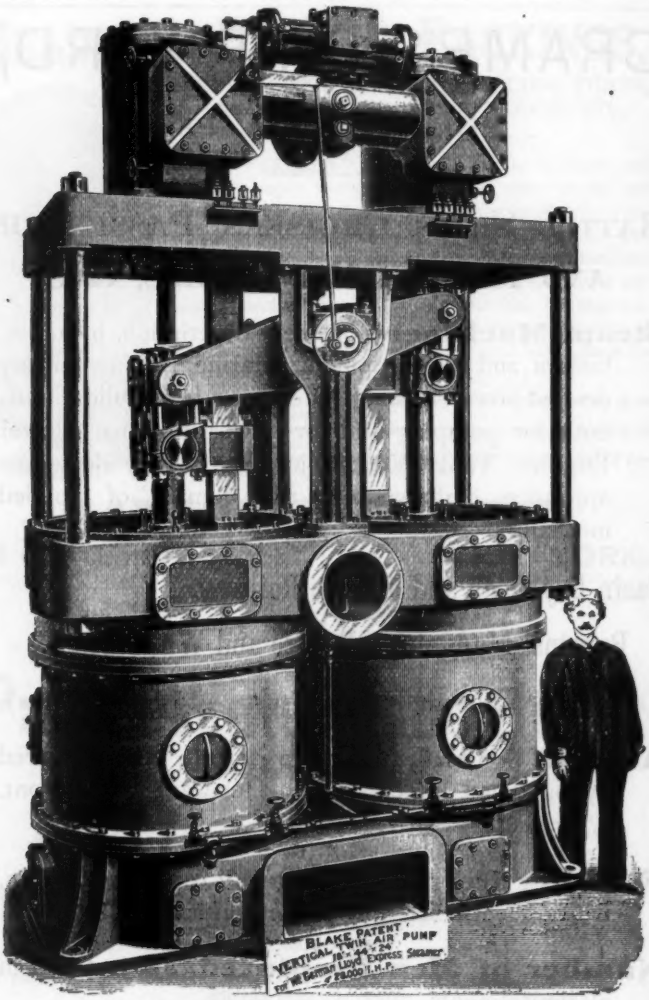

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Owing to unavoidable delay, the **General Index** will not be mailed until about June 10. The subscription list will close on June 1.